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Proceedings of 1978 Engineering Foundation Conference Waste Heat Utilization

August 13-18, 1978
New England College
Henniker, New Hampshire

U.S. Department of Energy
Assistant Secretary for Conservation
and Solar Applications
Office of Industrial Programs

Published: April 1979

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The Engineering Foundation of the
United Engineering Center

U.S. Department of Energy
Assistant Secretary for Conservation
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Washington, D.C. 20545

Published: April 1979

Forward

The Waste Heat Utilization Conference, sponsored by the Engineering Foundation of the United Engineering Center in August 1978 produced some discussions and papers of wide interest. In order to make these concepts, improved methods and equipment descriptions more widely available, the several papers have been collected and reprinted in this report. The subject, waste heat utilization, is one of vital national interest. Commercial use of the ideas discussed at the Conference, along with other improvements, can save, on a national basis, in the year 1985 over one quadrillion Btu's of scarce fuels.

The purpose of the Conference, as expressed by the sponsor, the Engineering Foundation, is to extend the frontiers of engineering by fostering free and informal exchanges among the knowledgeable and experienced participants, in an atmosphere that is free of the press of daily activity. They succeeded admirably. The format of the Conference, the accommodations and the physical surroundings were such that the conferee's were able, in their discussions, to define some of the gaps in the engineering knowledge, and indicate ways to fill these gaps.

The participation by the conferee's was frank and open and of a professional caliber. The Department of Energy's research and development program, that is being conducted in the national interest, will benefit from the conferee's efforts. We wish to thank them for their valuable assistance.

Douglas G. Harvey
Department of Energy

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COGENERATION AND OTHER INTERCHANGE
BETWEEN
INDUSTRIAL USERS AND REGULATED SUPPLIERS

BY

EDWARD V. SHERRY
AIR PRODUCTS AND CHEMICALS, INC.
ALLENTOWN, PENNSYLVANIA

Disparate competition between industry and utilities for steam and electricity supply investments is a cause of the relatively low industrial investments in self generation. Risk differences between utility and industry markets, and the need to conserve capital, are explored to explain the differing rate of return requirements for energy conservation and electrical generation investments. In the absence of production investments there is no opportunity for additional investments to improve energy efficiency. It is misleading to say that cogeneration does not occur merely because industry requires a higher return on such investments than on production investments. In some instances industry requires a lower rate of return on utilities for major plants. A broad definition of cogeneration is recommended as one of a list of 17 recommendations covering regulation, pollution, financial and tax policies.

ENGINEERING FOUNDATION CONFERENCE
NEW ENGLAND COLLEGE
HENNIKER, NEW HAMPSHIRE
AUGUST 14, 1978

Air Products

COGENERATION - PRICES & RISKS

The recent popularization of the word and concept of cogeneration has created the need for a thorough review of the nature of utilities and utility regulation, particularly the interface between monopoly areas of utility operation and areas in which there is competition. Disparate competition exists between regulated utilities and industrial firms to supply industrial plants with energy. This is particularly clear in the case of electricity supply. It is also true in the case of steam supply and is becoming increasingly apparent in the supply of substitute natural gas and low heating value gas. Competition also occurs in the opposite case where utilities vie with industry for investments to supply utilities with fuel, such as uranium mining and milling, as well as in other markets such as the petrochemical activities of the subsidiaries of major natural gas companies.

In the emerging re-examination of these relationships, the main focus has been on the differences in the steam and electricity markets and the detrimental effect that this has had on industrial generation of electricity.

Cogeneration

We are all familiar with the arguments--

POWER PLANTS
10,000 BTU/KWH
35% EFFICIENCY

COGENERATORS
5,000 BTU/KWH
65 TO 85% EFFICIENCY
250,000 LBS/HR STEAM
10 TO 30 MW OF POWER

- Utility power plants require about 10,000 HHV BTU to produce one kilowatthour of electricity while well balanced cogenerators require only 4100 to 5000 HHV BTU to produce a kilowatthour.

- Cogenerators operate at 65 - 85% efficiency compared to 35% for straight power production.

- A typical industrial plant that requires 250 thousand pounds per hour of steam can produce 10 to 30 megawatts of electricity depending on the top temperature and pressure and the process steam pressure level.

The benefits of increased industrial cogeneration of power appear so apparent and near at hand that there is a tendency to oversimplify; a tendency to believe that a few minor changes in regulations, or whatever, are all that is required, or more dangerous, a tendency to mandate cogeneration. A tendency to exaggerate the overall benefits and to rely on cogeneration as a cure all for the energy problem.

We need to return to basics and to admittedly oversimplified models and to build from these toward programs that will increase cogeneration.

The Electricity Market

The accepted basic model for the supply of electricity states that it is, by its nature, a monopolist's market. Neither individuals nor communities desire to duplicate or triplicate the high- and low-voltage delivery systems that would be required to keep two or three suppliers in the market. Economic theory suggests that economies of scale are so great that one supplier would become dominant in any given region, driving the competition out of business.

THE ELECTRICITY MARKET EXHIBITS LONG TERM STABILITY.

IN THE NATURE OF UTILITIES, MARKET RISK IS MINIMIZED.

Electricity is produced, delivered and consumed almost instantaneously in a tremendous diversity of uses by a vast variety of persons, corporate and individual. Because electricity costs are usually a modest portion of each consumer's income, and because electricity is generally considered irreplaceable, the market is stable. When one customer moves out, others move in; when one industry contracts, another expands. The nature of the system is such that market risk is minimal as long as even modest general growth exists.

We therefore accept the facts and grant the monopolist a franchise in return for the legal right to impose economic and other forms of regulation on his business. The regulation and franchise do not alter the basic facts of the monopoly, but both serve to reduce the already low business risks. The corresponding and consequent effect is that the overall financial returns on investment in utility supply are low.

With the flexible delivery system, the vast variety of customers, and the economies of scale, electric utilities can, and do, operate with high debt-to-equity ratios. They are comfortable with 35-year depreciable lives for major investments and sell common stock on the basis of a net income ranging from 9 to 11% of stockholders equity. The necessary addition of risk to obtain this return is provided by heavily leveraging the investment through a high ratio of debt-to-equity. The typical electric utility's net income as a percentage of assets is quite low, in the range of 3 to 5%.[1] The comparable figures for firms in general manufacturing are typically in a range of net income to assets of 2 to 11% but the range for net income to equity, although wider, is comparable to utilities at 9 to 20%. These statistics highlight the fact that the financial market makes the position of the stockholders of utilities roughly comparable to that of the stockholders of manufacturing firms - both groups have similar risks and similar returns. Despite the fact that the fundamental business of the utility involves less risk, the financial risk added by leveraging makes the circumstances comparable. This is important to keep in mind since it makes it very apparent that manufacturing managements would have no qualm about accepting the low basic returns on total investment that the utility receives so long as they could obtain the same low risk which would permit the same high leveraging. Conversely, if utilities were to invest in higher risk business, they must be permitted a higher return on this portion of the business.

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[1] Data from "50 Largest Utilities", Fortune, July 17, 1978 p. 127. Note that this is return on net assets - not return on rate base as used in utility regulation.

The Steam Market

STEAM SUPPLY BELONGS IN THE GENERAL MANUFACTURING REALM WITH ITS SET OF RISKS AND REWARDS AND DOES NOT FIT WELL WITH THE UTILITY BUSINESS.

In sharp contrast to electricity, steam can economically be sold only within a limited radius of up to 5 miles to a single or concentrated group of relatively large users. The market diversity is much smaller. A steam marketer and his customers share in the risks of the markets for the eventual products that are produced and sold to consumers. Economies of scale exist but are not as great as in electricity production. Reliability, standby and backup problems are severe in both cases but there are fewer solutions in the steam situation because of the absence of a large supply grid. Engineering variations of steam supply are greater because of the lack of ease of transformation to different pressures and corrosion and other problems related to both condensate return and the high percentage of raw makeup water to the boiler.

When all is considered, it is clear that steam supply belongs in the general manufacturing realm with its set of risks and rewards and does not fit well with the utility business.

The two business areas are fundamentally different. It is no suprise that the historical trend has been toward increasing separation. Industry tends to produce steam, utilities to produce electricity. The fundamentals of the two businesses make this the most economical route. Industrial steam facilities are often built for shorter economic lives and to serve limited markets. Utility power plants are built on a grand scale, to last more than a generation and to serve many different customers, through all sorts of technological and market changes that only marginally affect the electricity supply industry but could have major local effects on individual steam projects.

Return on Congeneration Investments

A STEAM PLANT MUST BE CONSIDERED IN A SIMILAR RISK CATEGORY AS THE INDUSTRIAL PLANTS WHICH IT SERVES.

COGENERATION EQUIPMENT SITTING ON TOP OF THIS PYRAMID OF RISK MUST CARRY THE COMPOUND RISK.

Why doesn't industry invest in added head pressure and higher top temperature equipment to produce electricity simultaneously with its steam requirement? Among other reasons, because such investments are very often clearly uneconomic for the reasons cited above.

The added investment would only be useful as long as the need for the base steam continues to exist. This need depends on the market for the products of the plant. Since a good portion of the steam use is in markets of high technological and other risk, the steam plant must be considered in a similar risk category. The cogeneration equipment sitting on top of this risk pyramid must carry the compounded risk. It is often, therefore, not in the best interest of society to have such an investment. The market system operates to prevent it through an electricity price from a diverse lower risk system, which will not justify the high additional return required by the industrial plant.

IT IS TOO MUCH OF AN OVER-SIMPLIFICATION AND MISLEADING TO SAY, AS MANY HAVE DONE, THAT COGENERATION IS NOT USED BECAUSE INDUSTRY REQUIRES A HIGHER RETURN ON CONSERVATION INVESTMENTS THAN ON DIRECT PRODUCTION INVESTMENTS. IN THE ABSENCE OF THE DIRECT PRODUCTION INVESTMENT THERE WOULD BE NO OPPORTUNITY FOR THE IMPROVEMENT ASSOCIATED WITH THE CONSERVATION INVESTMENT AND ECONOMICS DICTATE THAT RISK BE CONSIDERED IN THE OPTIMUM ALLOCATION OF CAPITAL.

It is too much of an oversimplification to say, as many have, [2] that cogeneration is not used because industry requires a higher return on conservation investments than on direct production investments. In the absence of the direct production investment, there would be no opportunity for the improvement associated with the conservation investment. Economics dictate that risk be considered in the optimum allocation of capital.

Utility Opposition to Industrial Generation

There is a natural tendency in any business to vigorously oppose competition. Electric utility monopolies are no exception; they can and have acted in a manner to prevent competition from manufacturers in the supply of electric power. Besides their justified advantages of scale economy and diversity, utilities tend to establish rate schedules and rules of practice and procedure that add additional burdens to potential cogenerators, especially when power production is excess to the cogenerators needs and must be sold to the grid.

---

[2] For example see, G. N. Hatsopoulos, E. P. Gyftopoulos, R. W. Sant, and T. F. Widmer, "Capital Investment to Save Energy" Harvard Business Review, March-April 1978, page 113. "This kind of situation often prevails in our economy. While utilities are making incremental investments in new facilities that yield only 8% on assets, energy users are turning down investments in conservation equipment (yielding an equivalent amount of electricity) if the projected annual return from the outlay falls below 30%."

UTILITY RESTRICTIONS
ON SELF GENERATION

HIGHER RACHETS AND MINIMUMS
HIGHER PRICES FOR BASE POWER SUPPLY
PROHIBITION ON CERTAIN RATE SCHEDULES
HIGH PRICE FOR BACK UP
LOW PRICE FOR SURPLUS POWER
RESTRICTIONS ON WHEELING
LACK OF INTER UTILITY COMPETITION

These restrictions take the form of unnecessarily complex regulations concerning parallel operation; higher ratchets, minimums and prices to industries that produce part of their own power; outright prohibition of any form of self-generation on some rate schedules; unrealistically high prices for backup and supplementary power and correspondingly low prices for surplus and export power; reluctance or absolute prohibition of 'wheeling' of power from one industrial site to another, either across the utilities transmission system or on private lines; the lack of facility for the industry to obtain offers from other utilities, near but not in, the franchised area, etc., etc., etc.

BY NATURE OF REGULATION, UTILITIES CANNOT MAKE MONEY FOR THEIR INVESTORS MERELY BY BUYING AND RESELLING A COMMODITY WITH A MINIMUM OF ADDED VALUE VIA INVESTMENT.

IT IS IN THEIR BEST INTEREST TO DISCOURAGE ANY FORM OF INDUSTRIAL POWER GENERATION.

We should not find such action surprising. Returning to the simplified case, the utility can be considered as two separate businesses. One operating at low risk with a 2 - 3% return on assets and the other managing the investment and obtaining an 11% return on equity. There is no opportunity for leveraging unless there is an investment. The utility, over the long haul, succeeds by making new investments. If it merely distributed and marketed power produced by manufacturing firms, it would have fewer investment opportunities. By the nature of regulation, utilities cannot make money for their investors merely by buying and reselling a commodity with a minimum of value added via investment. It is therefore in their best interest to discourage any form of industrial power generation.

The desire to make these investments has in some cases led utilities to ownership of cogeneration facilities, selling steam to industry and producing power for the grid. When such a facility can be built on a sufficiently large scale, selling to a diverse group of customers which limit the utility's risk via long-term contracts, the business approximates utility economics and makes it viable for utility ownership.

Industrial Ownership of Cogeneration

The majority of cogeneration possibilities do not, however, fit this mold. Under the ideal circumstances just outlined, industry would just as well invest in and own the entire cogeneration facility. It is wrong to say that in such a low risk case, protected by long term contract guarantees, industry would reject the low return that would be acceptable to a utility. Industry's only special requirement would be that the investment be 'off the books' via a separate joint venture or corporation in order to facilitate heavy debt financing and not interfere with the borrowing capacity of the base business. Industry may be especially prone to such efficiency improvement investments today when, due to overcapacity in many industries, profitable investment opportunities in new markets and expansions are more scarce.

MANUFACTURING MANAGEMENTS WOULD HAVE NO QUALM ABOUT ACCEPTING THE LOW RETURNS ON TOTAL INVESTMENT THAT THE UTILITY RECEIVES SO LONG AS THEY COULD OBTAIN THE SAME LOW RISK WHICH WOULD PERMIT THE SAME HIGH LEVERAGING.

Definition of Cogeneration

Not only the above 'classic' case, but other forms of cogeneration are better handled with industrial rather than utility ownership. In order to understand this, we need to broaden the definition of cogeneration. The definition used in early drafts of 1977-78 energy legislation, and later by the Department of Energy, has evolved from one which limited any special consideration to investments that simultaneously produce electricity and useful thermal energy to one which includes the sequential production of either electricity or shaft horsepower and useful thermal energy from the same primary energy source. When shaft horsepower is produced and utilized directly by compressors or other industrial equipment, utility investment becomes more difficult. If we broaden the definition of cogeneration to include production of electricity and mechanical energy using waste heat or low grade fuel sources such as industrial off-gases, liquid wastes, residuals from paper and other forest product production, municipal wastes, and even sources that have a negative energy content relative to ambient (such as liquefied natural gas) then we increase the potential for energy conservation and further reduce the potential for investment by utilities.[3]

COGENERATION IS THE SEQUENTIAL PRODUCTION OF ELECTRICAL OR MECHANICAL ENERGY AND USEFUL HEAT OR THE PRODUCTION OF USEFUL ENERGY FROM LOW GRADE 'WASTE' HEAT AND ENERGY SOURCES.

Bias Against Industrial Ownership

The original version of the Natural Energy Plan (NEP) tended to have a bias toward utility ownership of cogenerators, limiting the application of tax benefits and the advantages of special regulations to cogenerators owned by utilities or to facilties that were first offered to, but rejected by utilities. In order to obtain the maximum benefits of cogeneration, this concept should be reversed a full 180 degrees such that the emphasis is on a broad definition of cogeneration and on industrial ownership. Utility investment will then occur when it is feasible and economical.

---

[3] For a more complete list see E. V. Sherry, "Energy Conservation by Symbiosis", Energy, Summer/1977.

In the broadest sense, cogeneration, like load management, is any means of improving efficiency in the utilization of fuel and related capital. It can be argued that this definition is too broad but starting with broad concepts enables us to point out the diseconomies that may result from granting incentives to one specialized form of cogeneration while neglecting other beneficial methods. Once we have a broad definition we can refine it by defining subgroups. To aid in this analysis, the attached illustrations describe different types of cogeneration and load management to highlight both the similarities and differences.

FIGURE 1 shows a simplified utility power plant. Its characteristics, as contrasted with similar industrial equipment, are that it normally would be supercritical; use a small inventory of highly purified condensate; utilize a very expensive complicated cycle of reheats to obtain maximum efficiency; operate at a heat rate of about 9,000 Btu/KWH; exhibit strong scale economy up to at least 600 MW sizes; and take over six years to build.

FIGURE 2 shows cogeneration of the first kind. Heat requirements for industrial steam or district heating, for example, are provided by extract or back pressure steam from a turbine which is producing electricity. Heat rates in the range of 5,000 Btu/KWH can be obtained even with simpler boiler cycles and lower steam pressure, since the latent heat of condensation of steam that is lost to the condenser in a power plant is utilized for low level heat requirements.

We must consider three different cases of such cogeneration: Type A - utility owned; Type B - industry owned but producing only a part of that location's electrical needs and Type C - an industrially owned net exporter of power.

Utility owned cogeneration (Type A) has a long regulatory history, the plant of Gulf States Utilities at Baton Rouge is a prime example. Such projects are difficult to put together but have minimum regulatory problems. Type B is the case where an industrial plant generates power up to a balance with its steam requirement and purchases additional and/or backup power requirements from a utility. Types A and B probably account for the vast majority of practical applications of cogeneration. They are straight forward and should involve a minimum of regulation and price problems. Discussion of the other, more complicated cases should not be allowed to detract from maximizing these types.

Type C is cogeneration where the steam to power ratio of the industrial plant is high such that it would be efficient for the plant to export power. This is the type which raises the difficult questions of how to price export power. One extreme is to price it at system lambda, which is merely the out-of-pocket cost of the next increment of generation available to the utility. The other extreme is to set the price by analysis of the total cost of the industrial plant. A fair balance is to set the price at the market level. The same price at which the utility is selling power to other industrial customers (taking into account the time distribution, voltage of delivery, reliability and location of the source).

Setting the price at system lambda deters cogeneration, since the industrial company would have only the small margin resulting from the difference in its fuel efficiency and the utility's efficiency to defray the incremental cost of the added capital for the cogeneration equipment.

Pricing the power at market value would also have its problems. Utilities want to pay less for it during nights and weekends than at mid-day during the week. This problem would not exist if proper time-of-use rates were in effect; then both the selling and buying prices would reflect time differences in cost. If we set a limit on the amount of power that the utility would buy, based on the proper balance to steam usage, we would lessen the problem of setting up industry as a competitor to utilitiies.

Cogeneration at these prices would be an advantage to the national economy and would help reduce national fuel use, but would not be of obvious benefit to the utility or its other customers. To add utility incentive, our suggestion is the same as we made in 1970, in discussing load management techniques which reduce the need for utility investments. Utility rates of return should be on the high side of the zone of reasonableness if the utility can demonstrate that it is managing the system in a manner which promotes conservation and load management.

A second classification of cogeneration is the manufacture of electricity to optimize the use of industrial by-product heat, or by-product fuels, with or without simultaneous generation of steam, as illustrated in FIGURE 3. In this instance, utility ownership is infeasible. Type IIB would not present different problems than Type IB, but in Type IIC it would be more difficult to determine the quantity of electricity available for sale at market prices. The main objective in pointing out this type of cogeneration is to assure that it will be adequately recognized in any laws or regulations which gives incentive to cogeneration of the first kind.

FIGURE 4 shows cogeneration utilizing a gas turbine. Should such a plant, producing only electricity, be considered for cogeneration tax incentives since the first unit produces electricity and heat which is usefully put to work in the waste heat boiler?

Just as utilities would like to see the friendly aura conveyed by the word 'cogeneration' include this type of utility plant equipment, so would industry like to see it include similar normal energy conservative industrial practice. In many plants it is the practice to bypass the electrical generation step of cogeneration and use shaft horsepower from the turbine directly to drive process equipment. FIGURE 5, is an example which is common in process plants. Even small pumps are turbine driven by back pressure steam. The mere fact that industrial plants skip the electrical step, and the losses associated with the additional conversion, should not be used to indicate that they are not maximizing conservation techniques like cogeneration. National statistics on cogeneration should include production of shaft horsepower, not just electrical generation.

FIGURE 6 shows cogeneration for shaft horsepower, using gas turbines. An oxygen plant of this design operated in northern Indiana for many years.

Cogeneration is analogous to load management. The effect on the utility system of bringing on additional generation from an industrial source is similar to the effect of reduced use by an industrial customer. In either case, power is made available for others. The similarity of cogeneration and load management is illustrated by FIGURE 7. The equipment to the left of the dashed line is a normal air separation plant. Diverting some air to a combustor we have an air plant driven by a gas turbine. Oversizing the air compressor, we have a combination air separation plant and electrical generator which, of course, could supply added power at system peaks by load management and partial curtailment of the process operation.

Air Products

Policy Recommenations

If we are to retain private ownership and local management of utilities in the United States we must solve the problems of cooperation between regulated and nonregulated industries and between industry and government. In order to expedite this development, the following policy objectives are recommended to all parties involved:

PRIMARY RECOMMENDATION
BROAD DEFINITION OF COGENERATION
EMPHASIS ON INDUSTRIAL OWNERSHIP

1. Broaden the legal definition of 'cogeneration' such that special tax and legal treatment would apply to a variety of heat, steam, electrical and mechanical generation means that exhibit marked improvements over basic utility efficiencies in either the use of primary fuels or in the substitution of low value fuels and energy sources for primary fuels.

ELIMINATE ANTI-COMPETITIVE RESTRICTIONS
EXEMPTION FROM REGULATION
MODIFY STATE LAWS DEFINING UTILITIES

2. Eliminate anticompetitive restrictions to industrial self generation from utility policies.

3. Positive and specific statements need to be made in future federal laws and regulations to exempt cogenerators from utility regulation. It is insufficient to take a half-hearted approach by stating that exemption may be considered or exemption will be considered for one or more particular federal laws.

4. Change state laws that are interpreted to say that owners of electric generation equipment connected to the grid are public utilities.

5. Industrial cogenerators must understand their rights relative to use of the power they generate during emergencies due to local or national fuel, labor, or reliability problems. This is particularly critical today, since a major current industrial motivation for self generation is concern that utilities can no longer be relied upon as an adequate long term source of electrical energy.

PRICES SET BY:
LONG TERM CONTRACT
ARBITRATION BY PUC
BASED ON MARKET PRICE
MAKE WHEELING OF POWER ECONOMICAL

6. Establish regulatory procedures whereby the price for electricity sold, or power purchased, by an industrial cogenerator can be determined by negotiated contracts containing appropriate escalation and amendment provisions.

7. The price set by contract should be established relative to the market price for power at equivalent locations, voltage, reliability and time-related variations in supply. This price is obtained from the utilities' regulated cost of service and

not the industries' cost of production. Such a procedure preempts the motivation for governmental agencies to assume regulation of the industrial firms generating costs. In any event, the costs of by-product energy from industry are allocated joint costs of production of the main product and as such, are not amenable to cost of service type regulation. The prices so established would approximate a fair market price and would be based on the utilities' fuel costs which are higher than the cogenerators' fuel costs (due to the congenerators efficiency) and on the utilities' capital costs which are lower than those of the industrial plant (due to risk and economies of scale).

8. An arbitrator must sit in on the negotiation of these prices since it is in the utilities' best interest to set prices that would make the project nonviable and it is in the interest of other customers of the utility and national policy to obtain the highest price for supplementary power and the lowest price for surplus power as practical while assuring overall project viability.

9. Appropriate pricing provisions should be worked out for wheeling of power across utility systems and state lines from a cogenerator to other potential customers.

POLLUTION LAWS

ACCOUNT FOR EFFICIENCY

DO NOT BASE ON FUEL INPUT

MODIFY TO ACCOUNT FOR WASTE FUEL SITUATIONS

10. Pollution laws need to be modified to account for the higher emissions that occur because of the additional electricity or shaft horsepower produced. Regulations and laws based on energy input penalize efficiency improvements.

11. Pollution regulations based on primary fuels need to be modified so as not to discriminate against waste fuels such as forest residuals, municipal refuse, etc.

12. Pollution laws need to take into account the increased concentration of fuel consumption that will occur by cogeneration at the site of an industrial plant, or group of industrial plants, making a concession for the fact that the total fuel consumption is lowered.

TAX POLICY

IMPROVE DCF BY FASTER DEPRECIATION AND INVESTMENT TAX CREDIT.

DO NOT DISCRIMINATE AGAINST INDUSTRIAL OWNERSHIP IN TAX POLICY.

13. Industry should not be discriminated against compared to utilities through any tax incentive or disincentive in the allocation of fuels.

14. The discounted cash flow (DCF) for investments in qualified cogenerators, broadly defined, must be

improved by provisions for accelerated write-off, improved investment tax credit, relaxation of the 50% limitation of the investment tax credit, etc.

15. Include all of the equipment associated with the cogeneration investment in any tax concession, not simply the boiler or the turbine-generator.

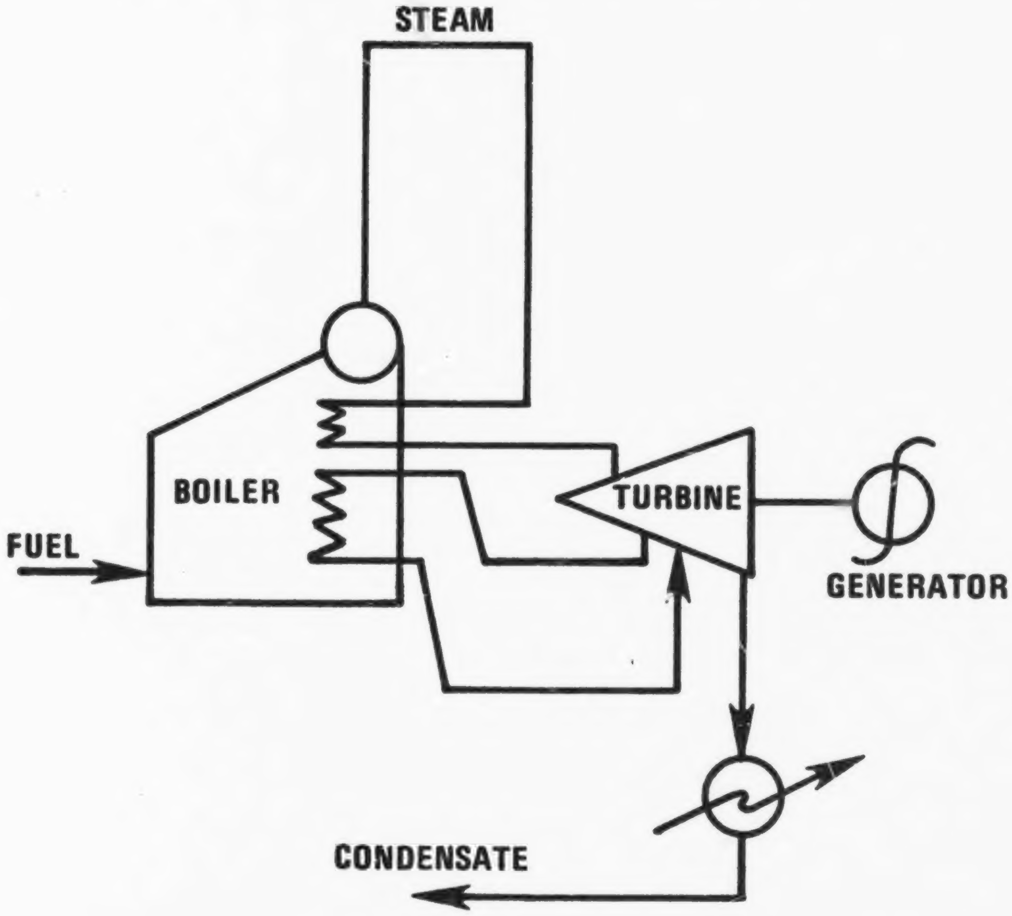
16. Qualified cogenerators must have access to oil supplies without tax penalties.

17. In the case of utility ownership of a cogeneration facility, the return on investment criteria need not necessarily be different than its other business since long-term contracts for the steam supply would minimize risk and aid in obtaining financing. If a project is not financable on this basis, the utility should be permitted to set up a separate entity for ownership and be permitted a higher return on investment commensurate with the risk it is willing to assume.

In order to expedite consideration of these issues, it would be desirable for the Department of Energy to use its influence to bring together a negotiating committee consisting of representatives of the Edison Electric Institute (EEI), the American Public Power Association (APPA), the National Association of Regulatory Commissioners (NARUC), the Federal Energy Regulatory Commission (FERC) and a comparable group of industrial consumers from organizations such as the Electricity Consumers Resource Council (ELCON), the National Association of Manufacturers, (NAM) and the Industrial Energy Group (IEG) to begin the establishment of the guidelines for cogeneration law and regulation.

FIGURE 1

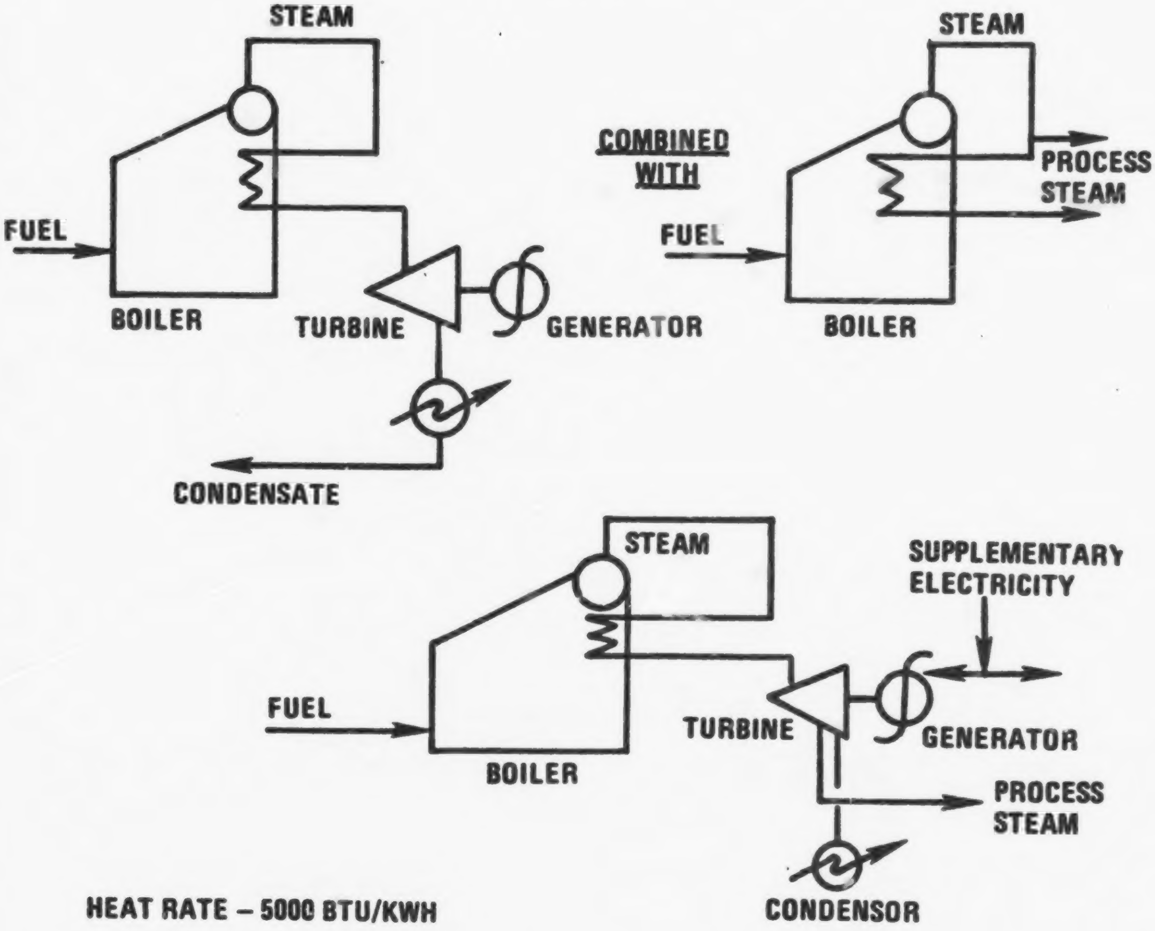
POWER PRODUCTION WITHOUT CO-GENERATION


STEAM
BOILER
TURBINE
FUEL
GENERATOR
CONDENSATE

HEAT RATE: 8-9000 BTU/KWH
ECONOMICAL SIZE: 600 MW
FUEL: COAL
LEAD TIME: >6 YEARS

FIGURE 2

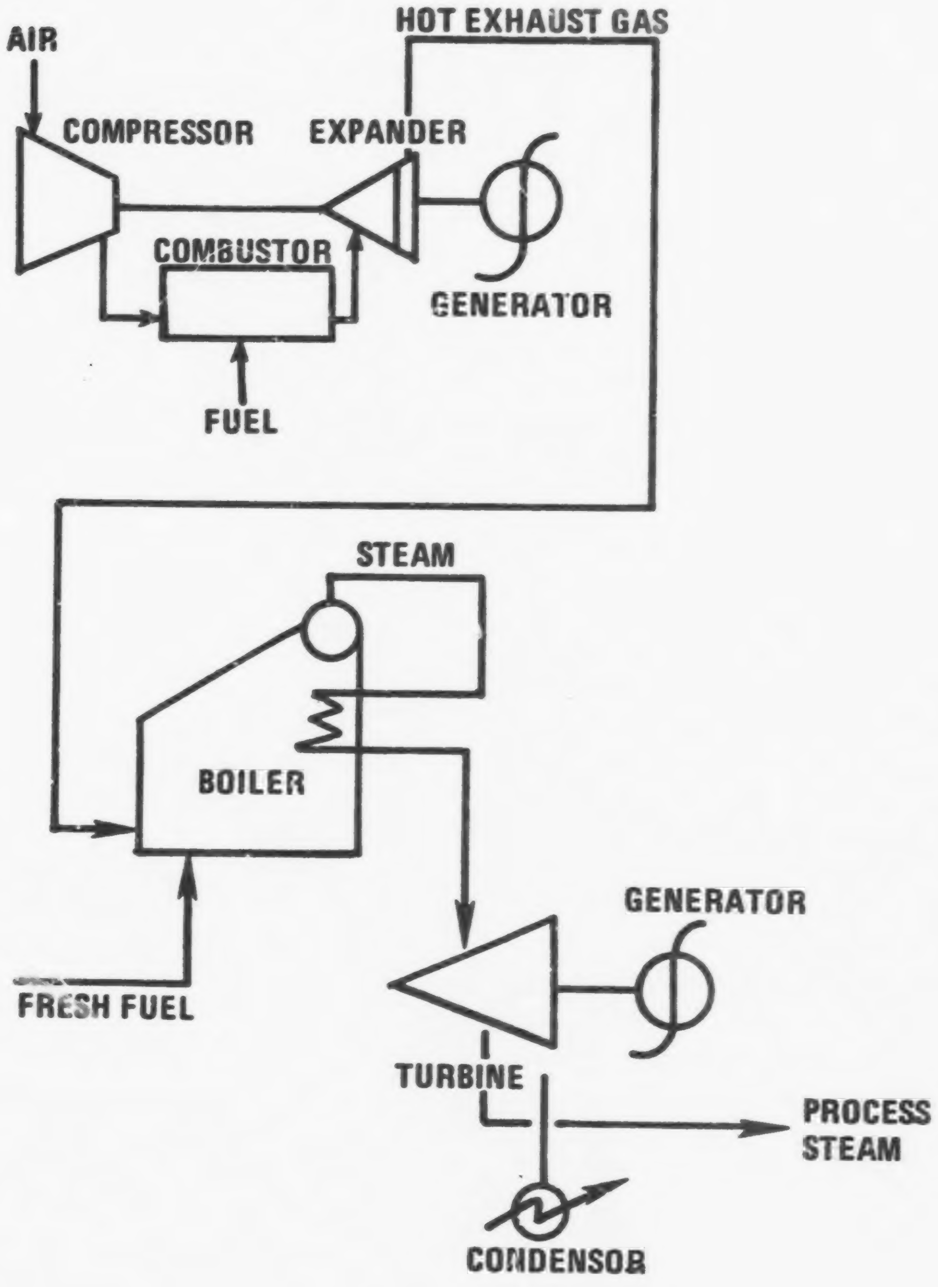
CO-GENERATION OF THE FIRST KIND


STEAM
FUEL
BOILER
TURBINE
GENERATOR
CONDENSATE
COMBINED WITH
STEAM
PROCESS STEAM
FUEL
BOILER
STEAM
SUPPLEMENTARY ELECTRICITY
FUEL
TURBINE
GENERATOR
BOILER
PROCESS STEAM
CONDENSOR

HEAT RATE – 5000 BTU/KWH

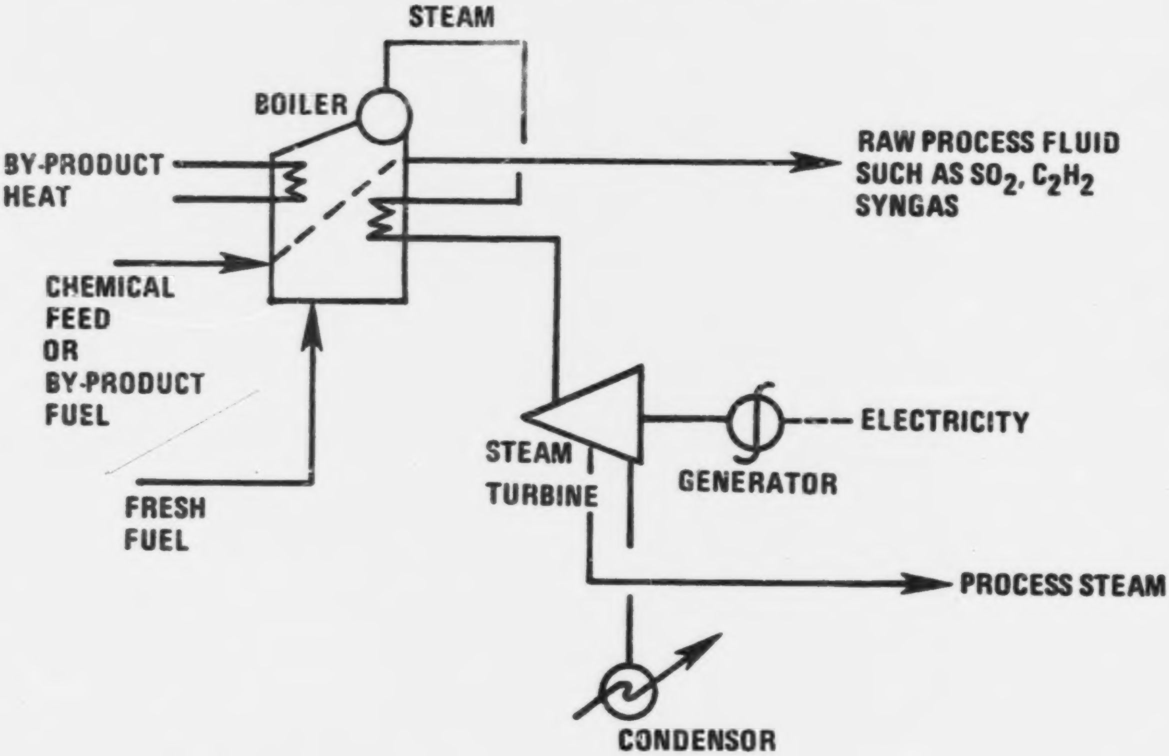
IA OWNED AND OPERATED BY UTILITY – SELLS STEAM AND ELECTRICITY
IB OWNED AND OPERATED BY INDUSTRY – PURCHASES ADDITIONAL POWER
IC OWNED AND OPERATED BY INDUSTRY – EXPORT POWER

FIRST AND SECOND KIND CO-GENERATION WITH HOT GAS TURBINES


HOT EXHAUST GAS
AIR
COMPRESSOR
EXPANDER
COMBUSTOR
GENERATOR
FUEL
STEAM
BOILER
GENERATOR
FRESH FUEL
TURBINE
PROCESS STEAM
CONDENSOR

CO-GENERATION OF THE SECOND KIND

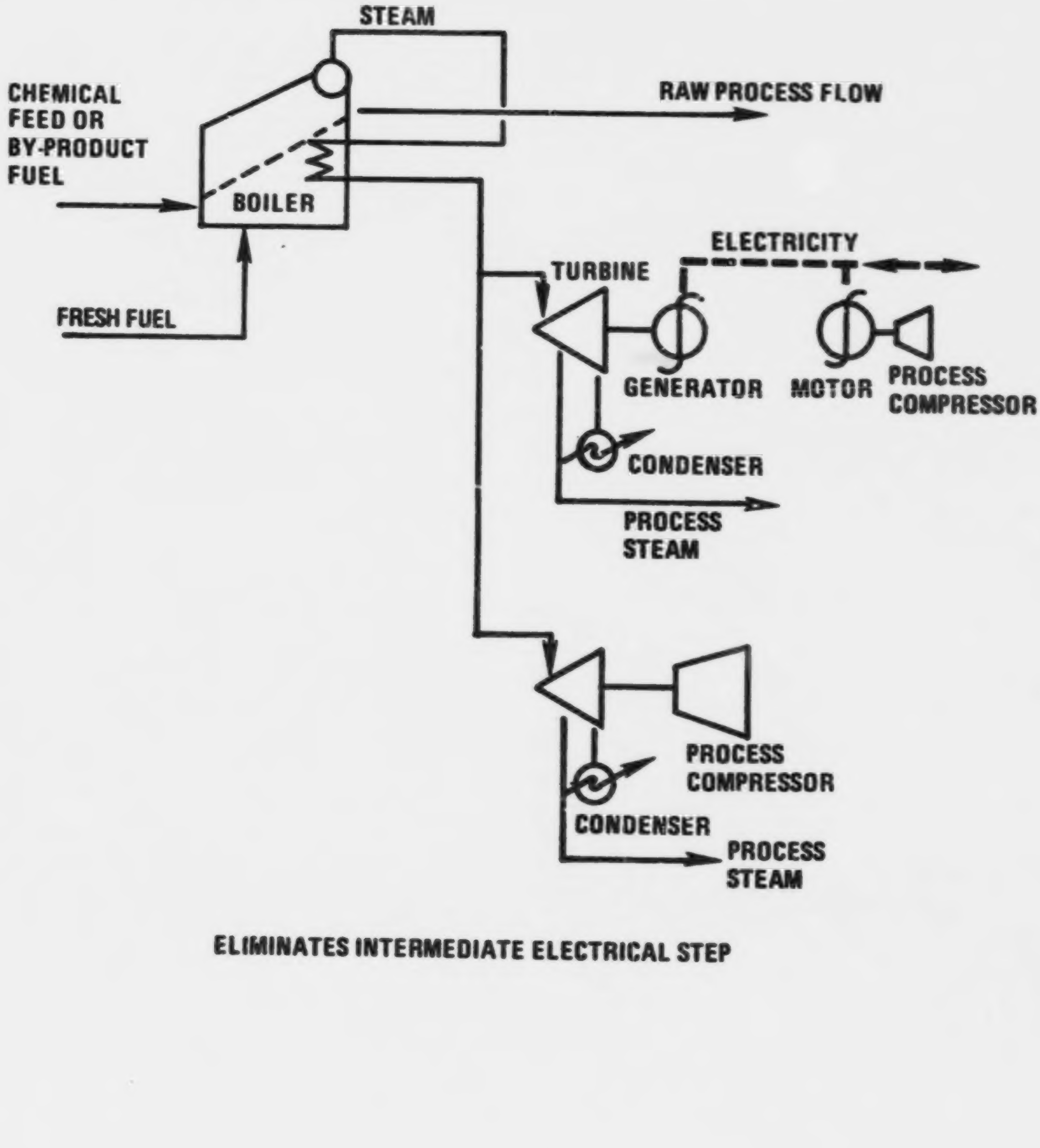
(BASED ON INDUSTRIAL BY-PRODUCT FUEL AND/OR HEAT)


STEAM
BOILER
BY-PRODUCT HEAT
RAW PROCESS FLUID SUCH AS SO_2, C_2H_2 SYNGAS
CHEMICAL FEED OR BY-PRODUCT FUEL
ELECTRICITY
STEAM TURBINE
GENERATOR
FRESH FUEL
PROCESS STEAM
CONDENSOR

IIA UTILITY OWNERSHIP IS NOT FEASIBLE
IIB OWNED AND OPERATED BY INDUSTRY – PURCHASES ADDITIONAL POWER
IIC OWNED AND OPERATED BY INDUSTRY – EXPORT POWER

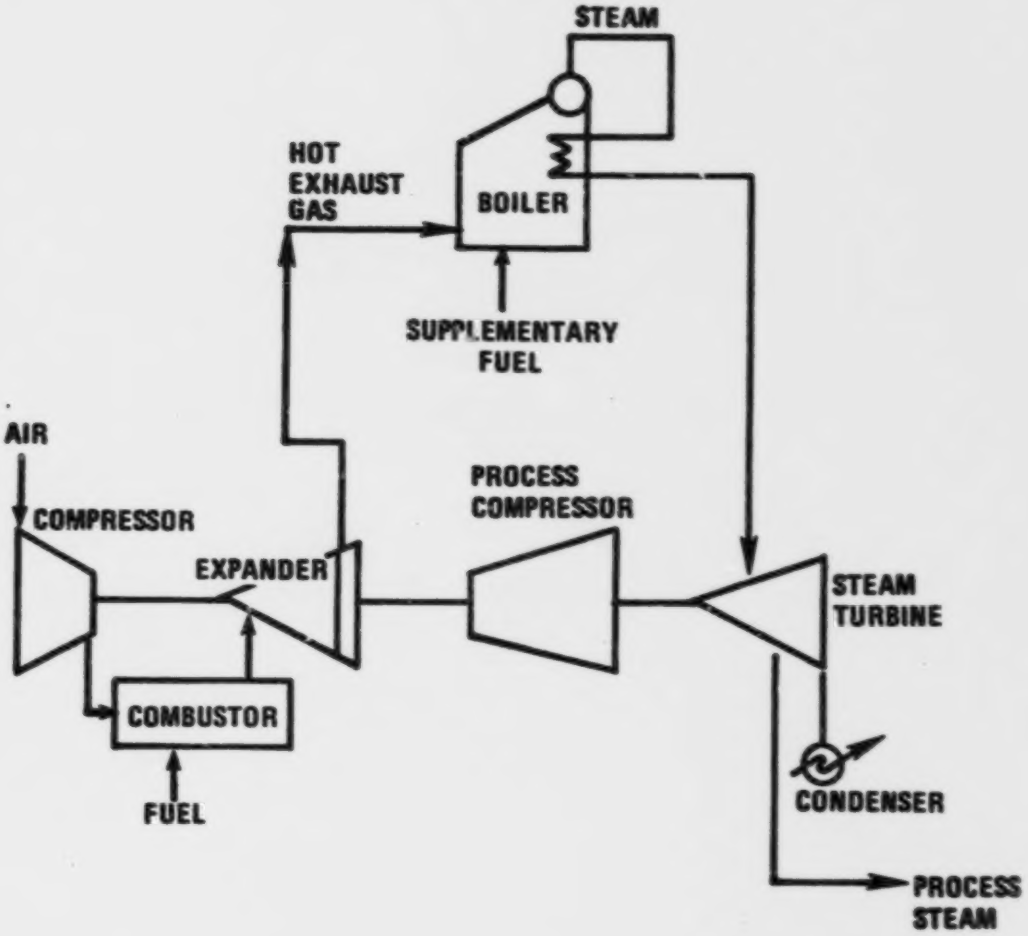
FIGURE 5

CO-GENERATION OF THE THIRD KIND

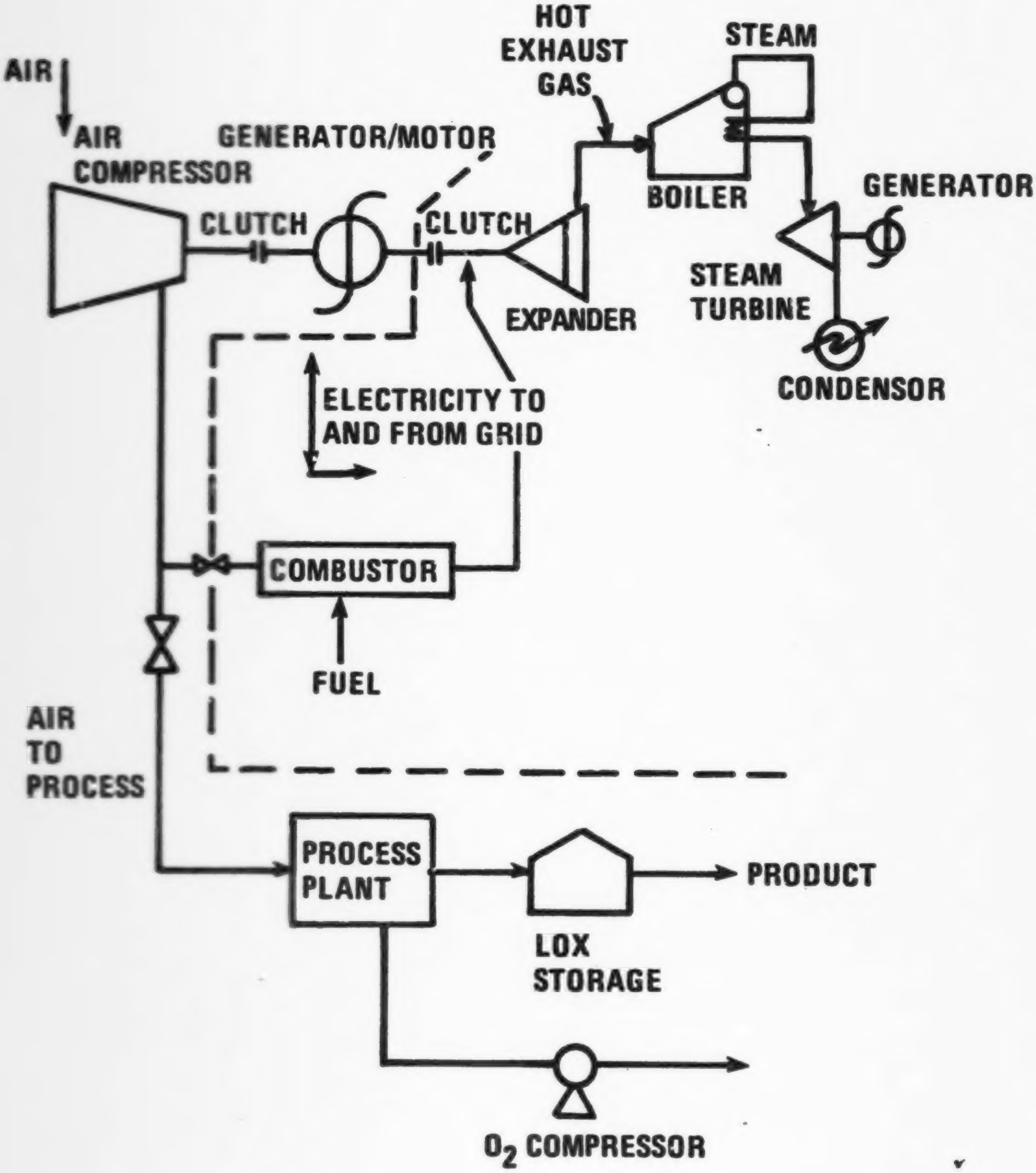

STEAM
CHEMICAL FEED OR BY-PRODUCT FUEL
RAW PROCESS FLOW
BOILER
FRESH FUEL
ELECTRICITY
TURBINE
GENERATOR
MOTOR
PROCESS COMPRESSOR
CONDENSER
PROCESS STEAM
PROCESS COMPRESSOR
CONDENSER
PROCESS STEAM

ELIMINATES INTERMEDIATE ELECTRICAL STEP

CO-GENERATION OF THE THIRD KIND WITH HOT GAS TURBINES


STEAM
HOT EXHAUST GAS
BOILER
SUPPLEMENTARY FUEL
AIR
COMPRESSOR
EXPANDER
PROCESS COMPRESSOR
STEAM TURBINE
COMBUSTOR
FUEL
CONDENSER
PROCESS STEAM

OPERATION OF AN AIR PLANT AT SYSTEM PEAK


HOT EXHAUST GAS
STEAM
AIR
AIR COMPRESSOR
GENERATOR/MOTOR
CLUTCH
CLUTCH
BOILER
GENERATOR
STEAM TURBINE
EXPANDER
CONDENSOR
ELECTRICITY TO AND FROM GRID
COMBUSTOR
FUEL
AIR TO PROCESS
PROCESS PLANT
PRODUCT
LOX STORAGE
O_2 COMPRESSOR

ENERGY MANAGEMENT SYSTEMS FOR COGENERATION

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ABSTRACT

Generating electric power along with process steam is not only economically attractive, but also is an essential ingredient in our national energy conservation program. Yet the performance of these generating facilities is entirely dependent on how carefully they are managed, particularly when one is faced with a wide range of electrical and steam loads, along with constraints imposed by equipment limitations and utility contracts. Most steam turbines and pressure regulators are designed to operate independently, and cannot be readily coordinated with other units. This paper describes the modifications necessary to accommodate the needed coordination, and the controls required to optimize performance of a typical multiturbine, multiheader facility. The economic and contractual factors associated with purchasing both steam and power are included in the solution.

INTRODUCTION

Cogeneration

Most processing plants have extensive uses for steam at several pressures. The lowest pressure steam is used for heating buildings and evaporators that operate under vacuum. Higher pressure steam is needed to distill mixtures under pressure, to dry products, and to separate high boiling components. Steam at still higher pressures is used to heat reactors and, in many cases, is injected directly into the reactors as in steam reforming processes.

The simplest means of supplying steam — and that which is most often used — is to install boilers having a rating that matches the highest pressure needs of the process. Shaft horsepower, also widely used in most plants, is generally provided by electric motors driven by purchased power. This arrangement is very clean and straightforward, but is costly to operate and wastes fuel.

The crux of the problem is this: steam has the capability of producing both work and heat, whose proportions are related to its state. A typical process plant needs both work and heat to function. But the typical central power station has a market only for work (electricity) and therefore must discard the heat it produces. Only about 32 percent of the fuel input to a central power station is converted to electricity; the balance is wasted. But since the process plant needs heat as well as work, it can use fuel much more efficiently by producing both.

Because the cost of fuels and electricity has risen dramatically in recent years, process plants are beginning to opt for higher pressure boilers to drive turbine generators. Exhaust steam from the turbines is then used for process heat. Steam is also being used to power rotating machinery through directly coupled turbines, thereby providing the added economy of variable speed operation.

Each process plant has its own peculiar requirements for both work and heat, and these requirements are subject to change with operating conditions, time of year, and other considerations. The proportions of work and heat obtainable from steam is a function of turbine inlet and outlet conditions which are also subject to change. As a result, it would only be coincidental if the work and heat requirements for the process happened to be matched by the work and heat availability of the steam being produced. An attempt may be made to balance them at the design stage of the plant, but even this is not likely to succeed. Moreover, conditions typically change so frequently that most plants -- once in operation -- are far out of balance most of the time. An intelligent energy management system is needed to manipulate turbines and pressure regulators to force the plant as close to a balance as possible, under the constraints imposed by loads, equipment, and contractural obligations.

A Typical Cogenerating Facility

Figure 1 describes a typical cogenerating facility, which will act as the vehicle to describe how an energy management system is to be applied. Shown are three turbogenerators supplying electricity to the process. Units 1 and 2 use 950 lb/in^2 steam produced on-site by power boilers; the third operates on 1200 lb/in^2 steam purchased from a nearby utility. Electricity may also be purchased from the utility, because the generation from all three turbines is expected to be insufficient to meet the demand for power.

Turbine 1 discharges into a condenser, with extraction going to the 50 lb/in^2 header. Turbine 2, which has no condensing stage, discharges into the 50 lb/in^2 header, with extraction at 250 lb/in^2. Turbine 3 discharges to the 250 lb/in^2 header alone. Also shown are two feedwater heaters, one of which is supplied by the 250 lb/in^2 exhaust header and the other by the 50 lb/in^2 header.

Waste-heat boilers are shown supplying steam to these exhaust headers. The production of these boilers is variable, since it depends on process conditions, and is not subject to manipulation. Production from the power boilers can be changed by adjusting their firing rates. To optimize plant performance, it is necessary to minimize the steam passing through the two pressure-reducing valves, because they bypass the turbines and allow steam to flow without doing any work. Most process plants need more work than they can obtain from their own steam, and therefore must purchase power. Consequently, these reducing valves must be kept closed under nearly all conditions. But if they are closed, they cannot be used to control pressure, which is their prime function. Plant optimization, therefore, requires another means of pressure control which does not waste work.

The electrical system also poses some interesting problems. If the system is independent of the power grid, then the frequency of the generated electricity must be regulated. Most turbines are equipped with a governor which manipulates the steam admission valve to hold speed (and hence frequency) constant. But most industrial plants are connected to the power grid, whose frequency is established by the many generating units on-line. Then the regulating function of the governors on individual turbines is eclipsed, and the generators will spin at the speed established by grid frequency, as if they were motors. The manipulation of a turbine admission valve is then no longer subject to its individual load, because the load is shared by a multiplicity of generators. Then the governor-equipped turbine cannot readily participate in a load sharing program of the type needed to optimally allocate steam. This situation is as much a problem as the role of the pressure-regulating valves described above. The usual practice is for the operator to manually distribute generator loading, while allowing the pressure-reducing valves to take up whatever slack remains.

A similar situation applies to the turbine readmission valves. Readmission of steam to the lower stages of an extraction turbine is typically accomplished under extraction pressure control. As extraction pressure rises, the readmission valve automatically opens. However, in most plants, extraction pressure is already regulated at the header. Consequently, the readmission valve will not change its position unless the operator manually intervenes. In essence, the stand-alone controls typically built into industrial turbines are immobilized when placed in a system already under control.

Contracts with utilities for the purchase of electricity, steam, and fuel, bear heavily on the economics of the cogeneration facility. Usually -- but not always -- the price of purchased electricity and steam will exceed the cost of that produced on-site. Furthermore, a limit on the demand for these services is customarily imposed by the utility. That limit could be a hard constraint which prevents any higher flow from being withdrawn; but more often, it is a soft constraint which imposes a cost penalty when flow exceeds the stated limit.

AN OPTIMAL ENERGY MANAGEMENT POLICY

The basic mode of operation for a cogeneration facility is to produce only the amount of steam at the lower pressure levels needed to meet process demands. Ideally, that steam should be supplied entirely from turbine exhausts. The demand for process steam appears as the output of the pressure controllers for the two headers. An increase in demand causes steam pressure to fall, and the output of the pressure controller to rise, so as to increase the supply. The problem is to find the optimum supply combination for each condition of load, equipment availability, constraint, etc. This is not an easy problem to solve, especially since nine control valves are available to control only two header pressures (see Figure 1).

There are two considerations which affect the implementation of an optimum policy: the limits of turbine capacity; and their dynamic responsiveness. Process steam load in a typical plant can vary widely and rapidly. However, a pressure-reducing valve can be made to move just as rapidly, to keep pace with the demand, and pass the load changes along to the boilers. It is questionable whether a turbine can or should change load as fast as needed to satisfy the process. Furthermore, a point will eventually be reached where the turbine admission valve has opened fully, and therefore, another source of supply is needed.

Header Pressure Control

The control system in Figure 2 has been devised to accommodate both aforementioned limitations at once. Header pressure is controlled by the reducing valve as is customary, but the turbine is manipulated so that it drives the reducing valve closed, in the steady state. This operation is accomplished by a valve position controller (VPC/N) acting on the signal to the reducing valve. If that valve is open above the valve-position set point, the controller will gradually increase turbine loading. This action will tend to raise header pressure, causing the pressure controller to close the reducing valve. The valve position controller operates on the integral mode alone, avoiding rapid upsets to the turbine and interaction with the pressure controller.

Implementation of the scheme shown in Figure 2 requires first that the turbine admission valve be disconnected from the governor, so that the valve is free to be manipulated by the valve position controller.

Should the demand for steam exceed the capability of the turbine to deliver it, the valve position controller (VPC/N) can no longer keep the reducing valve closed. The valve is then free to make up whatever additional steam is needed to satisfy the demand. The optimum operating range can be extended somewhat by closing off the valve sending steam to the feedwater heater. This is accomplished by a second valve position controller (VPC/H), whose set point is higher than that of VPC/N. The feedwater heater supply valve will not begin throttling then, until the turbine is fully loaded. The importance of this option is not great, because feedwater heating constitutes only a small fraction of the plant heat load. (To heat one pound of feedwater $100^{\circ}F$ only requires one-tenth of a pound of steam.)

Should a surplus of low-pressure steam develop, the reducing valve will close completely, and so will the turbine admission valve. The feedwater heater will, of course, be fully loaded. At this point, the readmission valve should be opened to supply the lower pressure or condensing stages of the turbine with the surplus steam. This operation is certainly preferable to wasting steam by venting or condensing it, or passing it on to a lower pressure header where it will create a surplus. The readmission valve is manipulated by another valve position controller (VPC/L), set below the set point of VPC/N. (To implement this scheme, the readmission valve must be disconnected from its accustomed pressure-regulating function. While this valve tends to perform the same service, even when not acting as a pressure regulator, it cannot be coordinated with other valves or manipulated for other purposes such as power demand control, which will be described later.)

To summarize, Figure 2 shows four control valves all coordinated to control header pressure in an optimum manner. In the normal range, the reducing valve position will be held at an average value, N, and the turbine will follow plant load. Should the turbine become fully loaded, the average reducing valve position will rise to set point H, and there the feedwater heater valve will begin to close. Further increases in load will be supplied by the reducing valve alone. Should the load fall so low that VPC/N closes the turbine admission valve, the reducing valve position will fall to set point L, where the readmission valve controls header pressure.

Determining Set Points

In the ideal situation, the reducing valve would be closed, the feedwater valve open, the readmission valve closed, and the admission valve throttling. Rapid load changes cannot be accommodated by the turbine, however, as established by the integral time of VPC/N. If the reducing valve is held closed, then, only increases in steam demand can be followed rapidly, with decreases causing header pressure to rise. Furthermore, pressure control is likely to be erratic when the reducing valve is operating at its seat, even in the absence of load fluctuations.

If pressure control is to be retained in the unsteady state, the position of the reducing valve must be above zero for all reasonable load excursions. This is accomplished by an "intelligent" valve position controller that generates a set point internally as a function of the amplitude of the fluctuations in valve position. In essence, the controller manipulates the turbine admission valve so that the minimum position of the reducing valve will approach zero, regardless of the magnitude or frequency of its excursions, as described in Figure 3.

The VPC has a local set point which serves as the base or minimum value to which valve position is to be controlled. For VPC/L this would be set at zero; VPC/N should be set slightly above zero, e.g., 2 percent; VPC/H would be set at perhaps 10 percent. All three controllers would respond to fluctuations in reducing valve position in the same way, i.e., higher amplitudes would precess all three internal set points upward by the same amount. In this way, their locally-set bias to one another would be retained as in the steady state, and only one of the three would be controlling at any point in time.

Coordinating Turbines 1 and 2

Exhaust steam is supplied to the 50 lb/in^2 header from both turbines 1 and 2. Then VPC/N for that header may manipulate both the admission valve for turbine 1 and the readmission valve for turbine 2 in parallel, as shown in Figure 4. This signal is combined with the output of VPC/L for the higher pressure header, by means of a high selector. Then the readmission valve may open to relieve a surplus of steam which will reduce the demand at the 50 lb/in^2 header, and cause turbine 1 to throttle back.

Should the demand for steam at the 250 lb/in^2 header be higher than the turbines can supply, VPC/H will override the readmission valve, as it closes the valve to the feedwater heater. This is accomplished through a low selector following the high selector. Under normal conditions, the feedwater heater valve would be fully open; therefore the low selector would not interfere with turbine operation.

The signal downstream of the low selector is fed back to the integral mode of VPC/L to avoid windup in case of an override. The same protection is not required for the other controllers, because they are also manipulating other valves at the same time.

Optimum Header Pressures

Header pressures should be maintained as low as possible, while still satisfying all users of steam from those headers. If all user valves are throttled, then header pressures are higher than necessary, and the valves are taking a pressure drop that would more profitably be applied across the turbines. Furthermore, reduction in header pressures will cause an increase in temperature difference across the heat-transfer surfaces in waste-heat boilers connected to them. This will allow the recovery of more waste heat than otherwise possible.

Optimum header pressure can be attained by examining the valve positions of users as in Figure 5. The valve position controller attempts to hold the most-open valve near its fully open position by adjusting the set point of the pressure controller. This must be done slowly, through integral action, to avoid upsetting the other steam users. Here the valve-position set point is internally generated to keep the peak points of variations in position near 100 percent of scale.

When there are a great many users, this approach becomes unwieldy to implement and is unlikely to achieve worthwhile gains. Then the engineer must look for alternate strategies. There may be a single user or group of users more limited than the rest and hence they can be chosen to index the pressure set point. Or an operator may be able to adjust the set point seasonally, or based on whether certain plant units are in operation. In any case, there is an optimum set point for each set of conditions, and some strategy should be investigated to determine and follow it.

ACCOMMODATING ECONOMIC FACTORS

Saving energy is not always consistent with saving dollars, particularly when contracts with utilities are involved. Normally, purchased power is more costly than that generated on-site, but there may be off-peak intervals where the reverse is true. The same could also apply to purchased steam or gas.

Another factor is the cost penalty for exceeding a contractual limit on the rate of use of power, steam, or gas. Any energy management system must automatically honor these limits, and, if necessary, must employ more costly resources to satisfy the peak demand.

A third factor that must be accommodated is the capacity and availability of equipment. When a turbine supplied with less costly steam is fully loaded, any additional supply will have to be drawn from another turbine, using more costly steam. The possibility also exists of the first turbine being shut down for repairs, or having an insufficient steam supply due to a boiler outage. An effective management system must be able to account for all these factors and to transfer control smoothly from one valve to the next as the situation changes.

Power Demand Control

Under ordinary circumstances, condensing stages of industrial power generating turbines should not be used. In principle, this is how the utilities operate their turbines, and a smaller cogeneration facility simply cannot compete with them at their own game. Normally, then, the valve readmitting steam to a condensing section would be resting at its minimum position, and would pass only enough steam to cool the blades.

Should a surplus of steam develop at the header feeding the condensing section, the readmission valve should then be opened by VPC/L, as described earlier. But, if purchased power reaches the demand limit established in the contract, the condensing stage should be used as well. These functions are combined by the control system shown in Figure 6. A high-signal selector will open the valve on command either of VPC/L or the kilowatt controller (KWC). A common feedback signal sent to both controllers allows manipulation of the valve to transfer smoothly from one to the other when both measurements are at their respective set points. If the kilowatt controller cannot be satisfied, and drives the readmission valve fully open, an alarm on its output signal can be applied to alert to the need to initiate a load-shedding program. Note that this alarm is not actuated by VPC/L.

A summer has been inserted in the signal to turbine 1 admission valve, to increase its opening upon demand from the KWC. The reasoning here is that opening of the readmission valve by the KWC will increase the demand for steam at the low-pressure header, which the admission valve must supply. However, this input is not activated by VPC/L, because when the latter is controlling, there is no demand for steam from the turbine.

Coordinating Turbines 2 and 3

This problem involves consideration of steam cost and availability. Steam produced on-site ought to cost less than that purchased, but allowance must be made for the reverse situation. The lead turbine should use the less costly steam up to the point of maximum loading, where control is to be transfered to the following unit using more costly steam. But it is important that control be equally effective, regardless of which turbine is taking the load swings, and that transfer between them be smooth. Furthermore, limits on boiler and turbine capacity must be accommodated as well ... again without compromising control.

The system shown in Figure 7 has been devised to meet these multiple objectives for the case of two different sources of steam. The operator selects the turbine using the less costly steam to take the load. The summer calculates the lead turbine valve signal, without any negative bias from the other turbine, so that the admission valve signal is directly proportional to the output of VPC/N. The gain k of its positive input is set to drive the admission valve full stroke when the output of VPC/N reaches a value equal to that turbine's portion of their combined capacity. For example, if turbine 2 were rated at 100,000 lb/h of steam and turbine 3 at 150,000 lb/h, lead turbine 2 should reach full loading at 40 percent output from VPC/N. Then k_2 = 250/100 or 2.5, and k_3 would be 250/150 or 1.6 percent.

Figure 7 shows turbine 2 selected to lead. Then the summer for following turbine 3 sees k_3m at its positive input and $(k_3/k_2)v_2$ at its negative input; m is the output of VPC/N and v_2 is the position of the valve admitting steam to turbine 2. If there is no limit applied to turbine 2, then $v_2 = k_2m$, and $v_3 = k_3m - (k_3/k_2)v_2 = 0$. Thus the following turbine will remain at zero load. Should turbine 2 reach a high limit imposed by tracking controller HIC/2 or the subtractor downstream, v_2 may fall below k_2m, causing turbine 3 to pick up the difference in load. (Tracking controller HIC/2 has its output connected to its measurement input and is normally in automatic, so that output follows input.)

Under normal boiler operation, the high-pressure header is controlled by setting total fuel flow to all boilers. Should the fuel flow systems be unable to meet the demand, a sustained deviation will appear across the total fuel controller (ΣFC). This same deviation is applied as a negative bias to reduce the control signal to turbine 2, and bring about a corresponding increase in loading to turbine 3. At the same time, an alarm acting on the deviation signal, places the tracking controller HIC/2 in manual, causing it to hold its last output, and prohibiting any further manipulation of turbine 2 by VPC/N. This effectively transfers control of reducing valve position to turbine 3, and places turbine 2 under control of the high-pressure header. A lag in the deviation signal to turbine 2 simulates the lag in the firing circuit, which the high-pressure controller normally encounters. An adjustable gain G on the negative bias to turbine 2 allows the pressure control loop to be properly damped while the turbine is being manipulated.

When the deviation returns to zero, either due to a reduction in load or recovery of the fuel system, HIC/2 will be returned to automatic, driving turbine 2 toward the loading established by VPC/N, at the integral time set in HIC/2. Both HIC stations are equipped with output limiters to be set at the maximum desirable turbine loading. This limit not only stops the lead turbine from increasing its loading, but also activates the following turbine. The lead turbine may also be "base loaded" with its HIC in manual, and the following turbine will automatically take the balance of the load.

CONCLUSIONS

The steam distribution system for this typical cogeneration facility has been shown to be optimized by the application of conventional process controls. Implementation may be accomplished with analog pneumatic or electronic instrumentation. A digital computer is unnecessary. The only decision the operator must make is the selection of a lead turbine, based on the relative costs of his two sources of steam. All other functions are automatic, and are accomplished in real-time. Control loop overrides and transfers take place instantaneously and smoothly, to follow changing plant loads.

Although plants and their requirements differ, the combinations of valve position control with variable loop structuring is applicable to all.

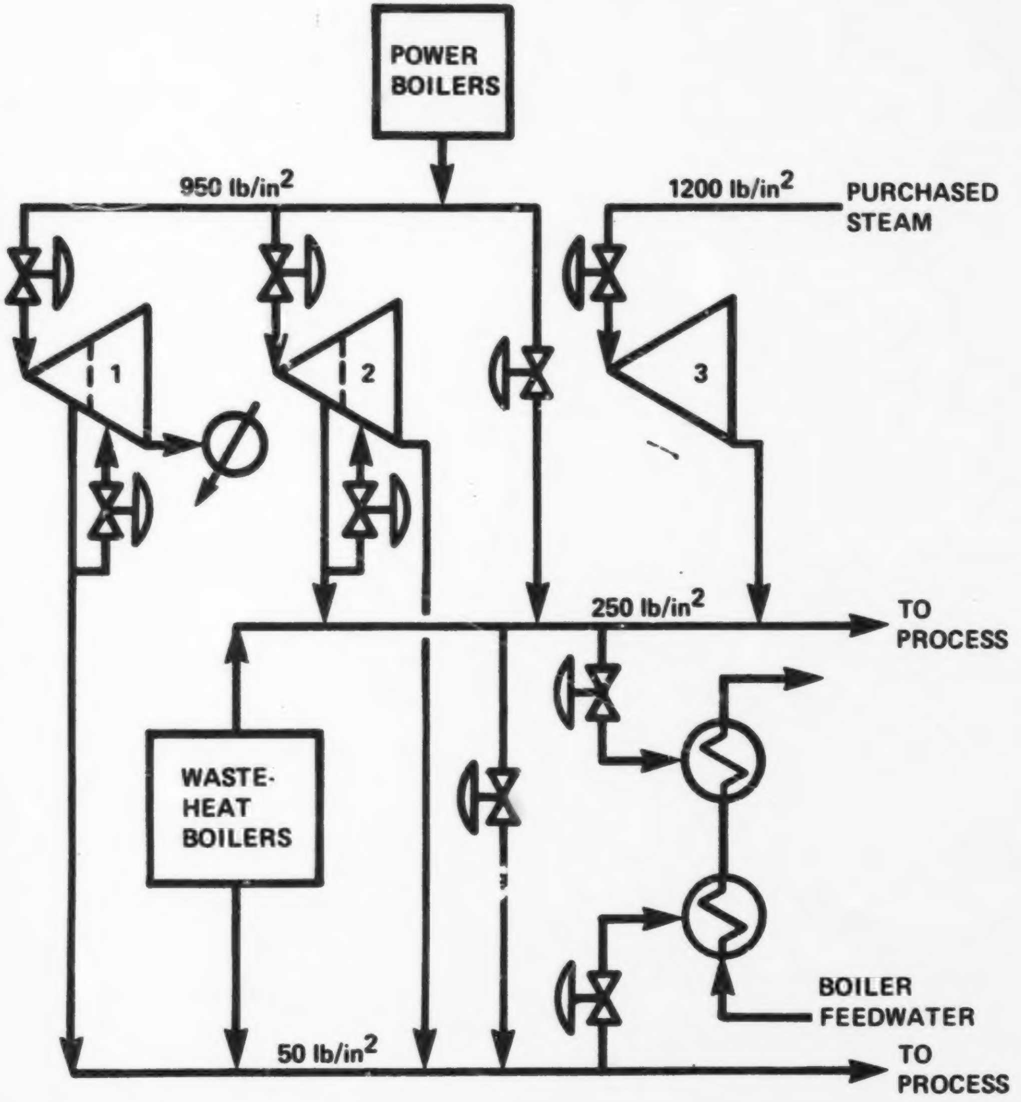

POWER BOILERS
950 lb/in^2
1200 lb/in^2
PURCHASED STEAM
1
2
3
250 lb/in^2
TO PROCESS
WASTE-HEAT BOILERS
BOILER FEEDWATER
50 lb/in^2
TO PROCESS

FIGURE 1. A typical cogeneration facility has several turbogenerators discharging into two or more steam headers. In this example, note that nine valves are available to control only two header pressures.

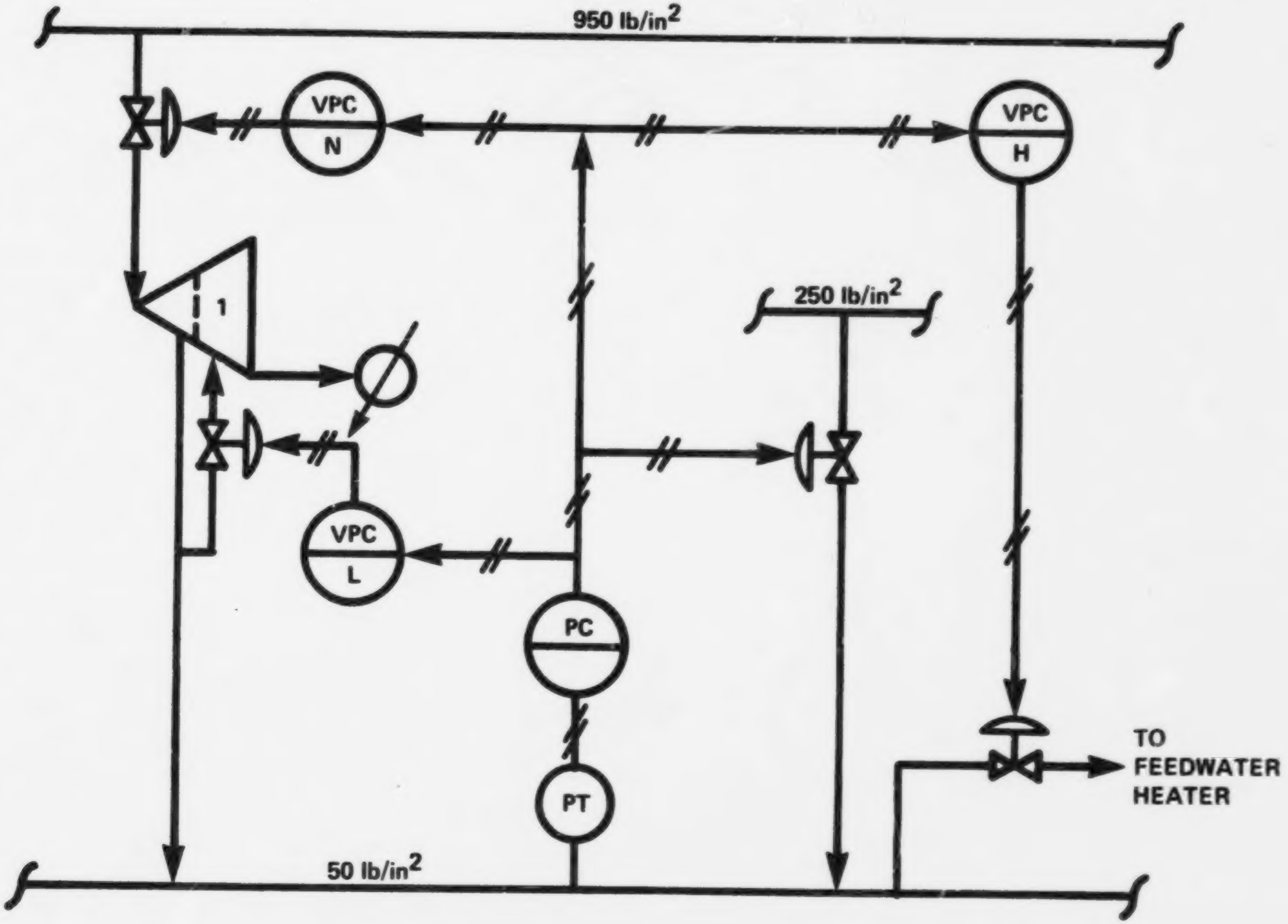

950 lb/in²
VPC
N
VPC
H
1
250 lb/in²
VPC
L
PC
PT
TO FEEDWATER HEATER
50 lb/in²

FIGURE 2. The valve position controllers pass sustained load variations along to the admission and heater valves, in order of priority.

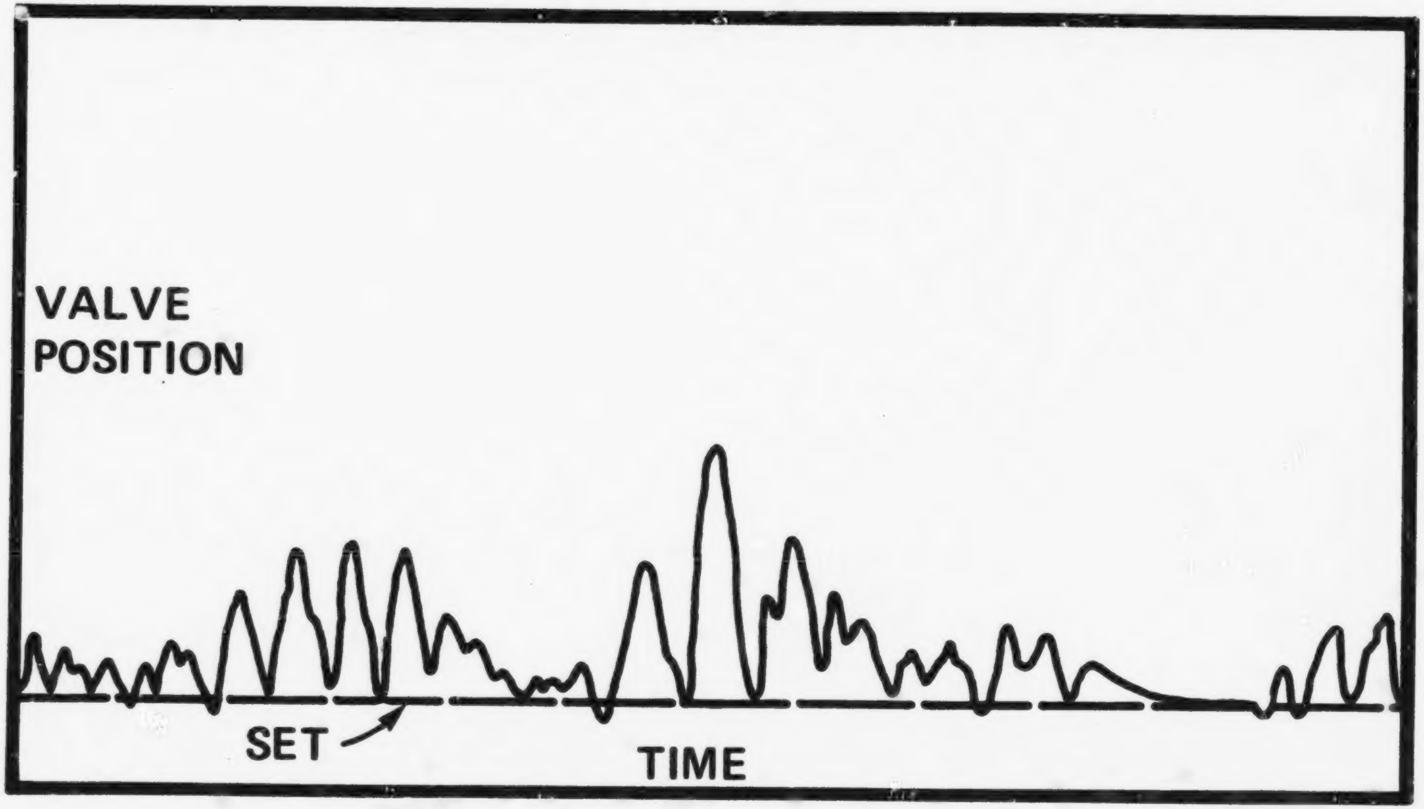


FIGURE 3. The "intelligent" valve position controller tries to bring the minima of a series of excursions to the set point.

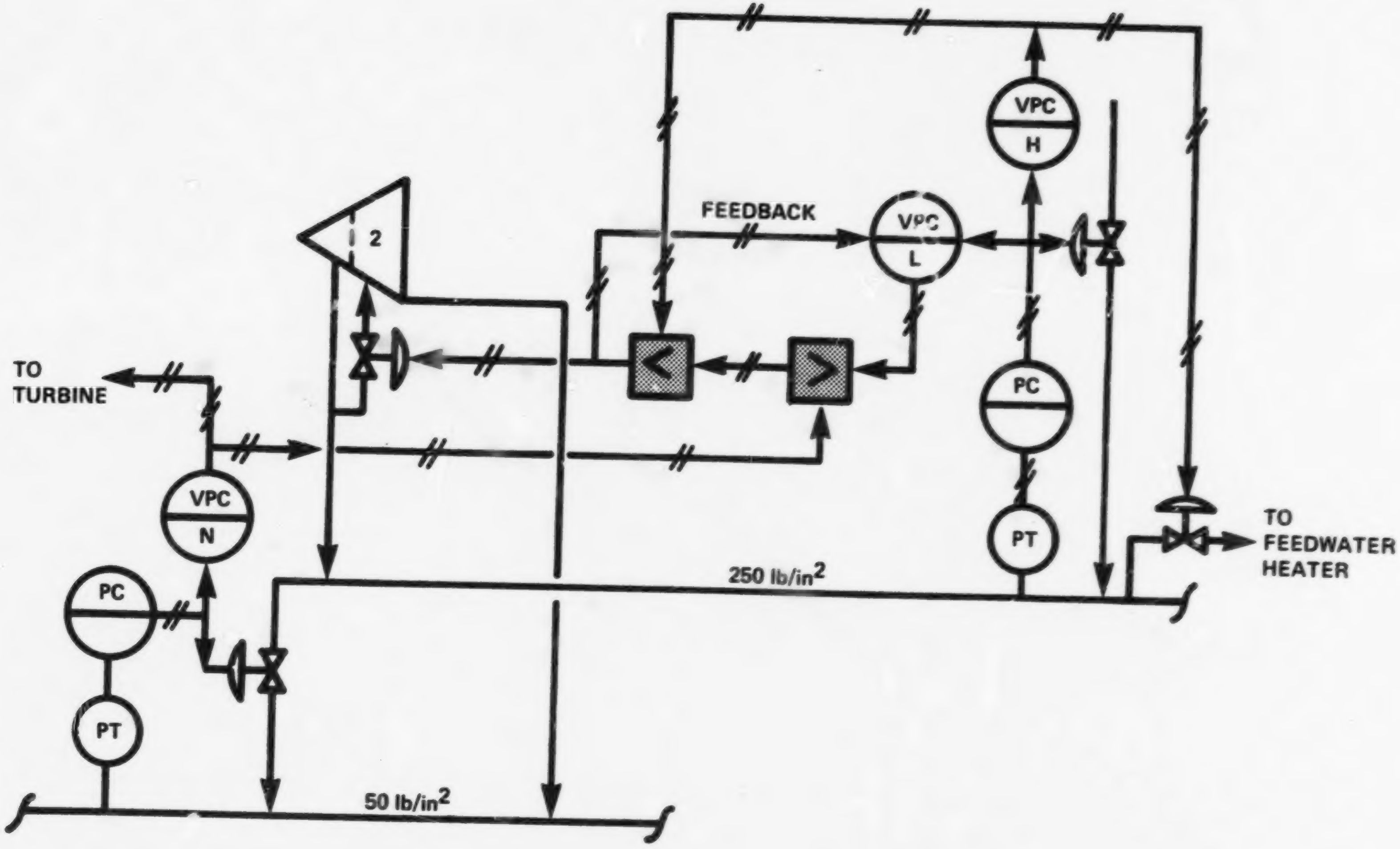
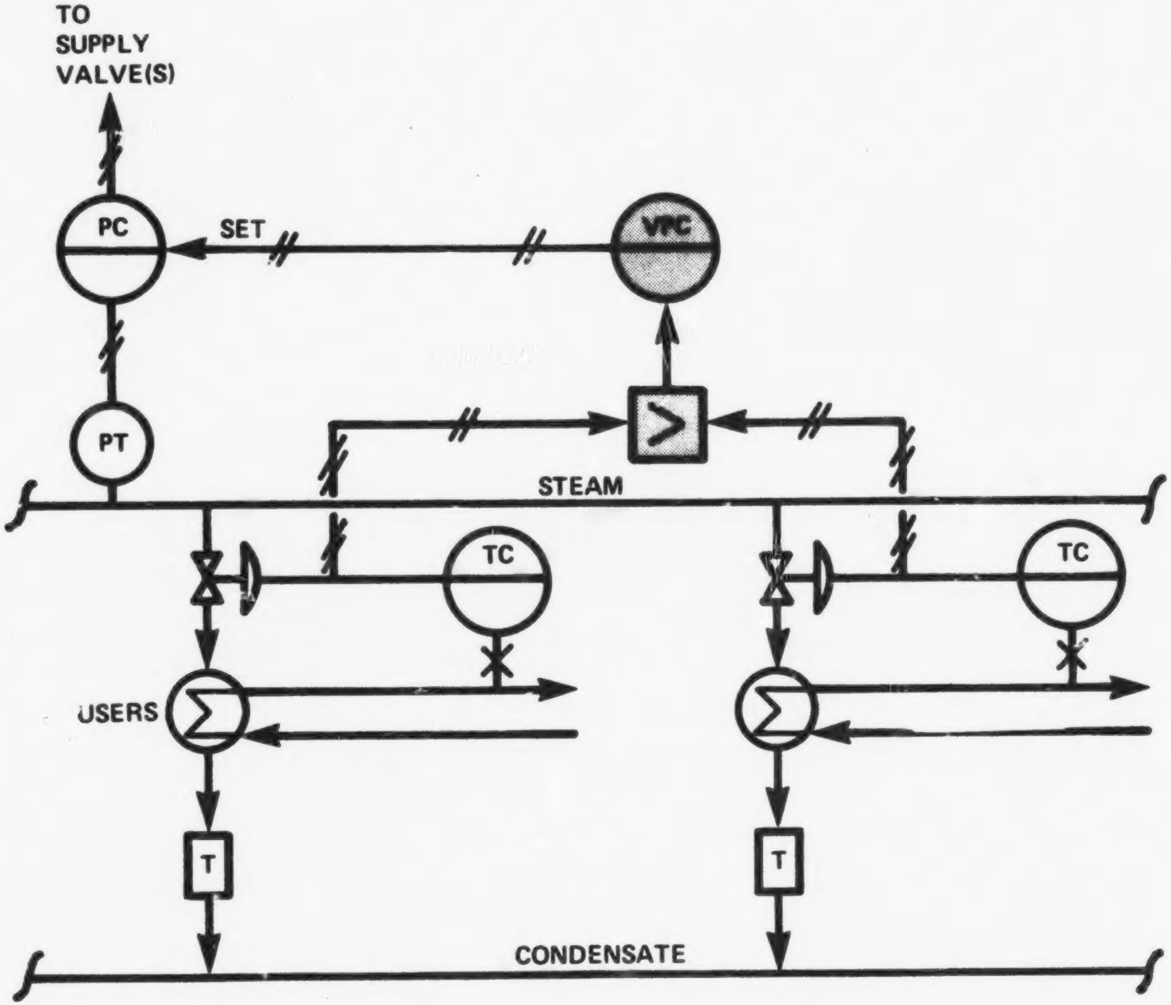

VPC
H
FEEDBACK
VPC
L
2
TO
TURBINE
<
>
PC
VPC
N
PC
PT
PT
TO
FEEDWATER
HEATER
250 lb/in2
50 lb/in2

FIGURE 4. Turbine 2 readmission valve may open to add steam to the 50 lb/in^2 header or to relieve steam from the 250 lb/in^2, but VPC/H closes it if the demand for steam is too high. (Note: added components are shown shaded.)



TO
SUPPLY
VALVE(S)
PC
SET
VPC
PT
STEAM
TC
TC
USERS
T
T
CONDENSATE

FIGURE 5. The valve position controller will gradually lower steam pressure until the most-open user valve approaches 100 percent open. (Note: added components are shown shaded.)

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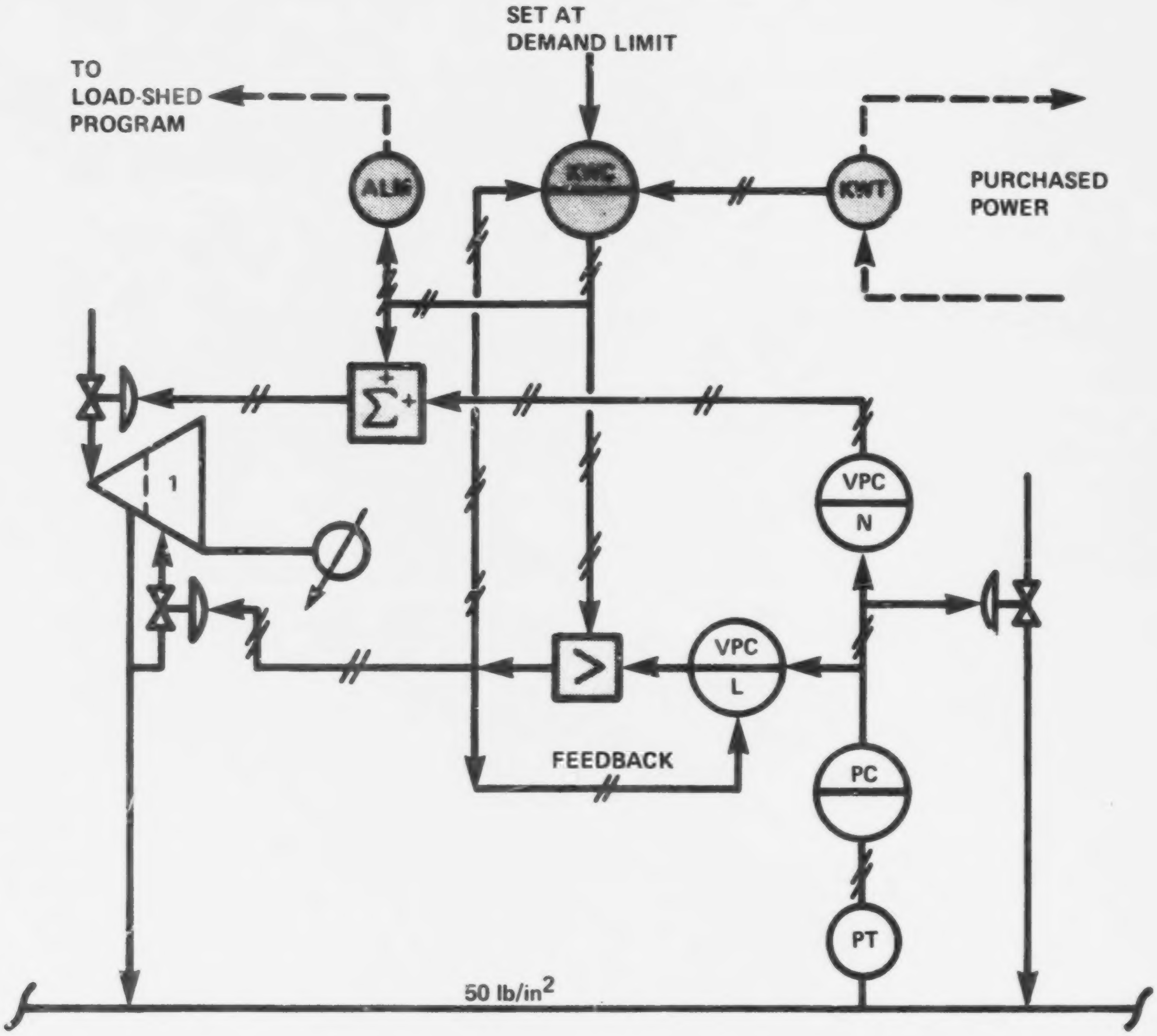
SET AT
DEMAND LIMIT
TO
LOAD-SHED
PROGRAM
ALM
KWC
KWT
PURCHASED
POWER
Σ
1
VPC
N
VPC
L
>
FEEDBACK
PC
PT
50 lb/in²

FIGURE 6. The kilowatt controller opens the [illegible] ondensing stage only when the demand limit is reached. (Note: added components are shown shaded.)

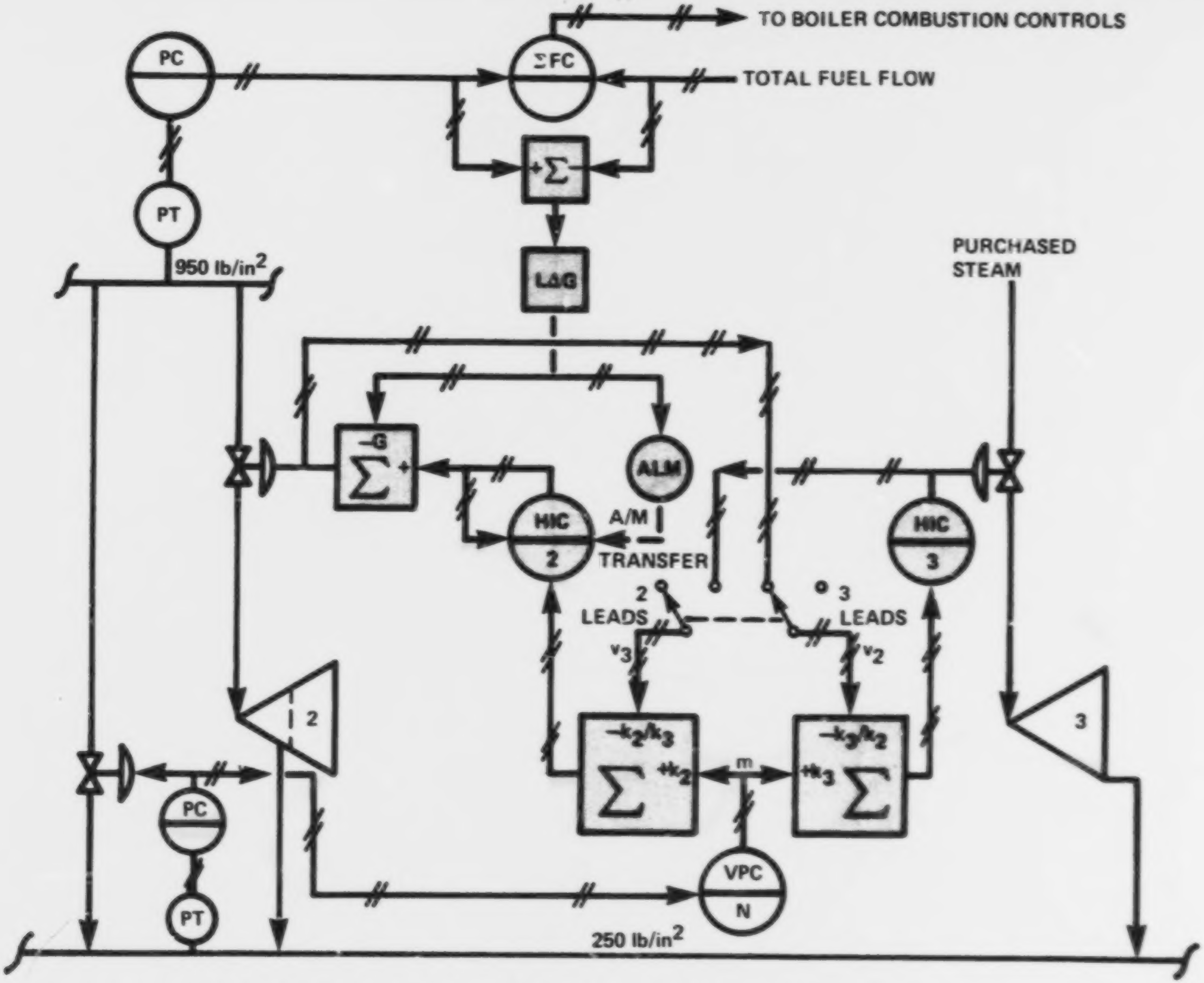
TO BOILER COMBUSTION CONTROLS
TOTAL FUEL FLOW
PC
ΣFC
+Σ
PT
950 lb/in²
LAG
PURCHASED STEAM
−G
Σ
+
ALM
HIC
2
A/M
TRANSFER
HIC
3
2 LEADS
3 LEADS
v3
v2
2
3
−k2/k3
+k2
m
−k3/k2
+k3
PC
PT
VPC
N
250 lb/in²

FIGURE 7. The turbine using less costly steam is selected to lead. The high-pressure controller automatically overrides turbine 2 in case a boiler capacity limit is encountered.

HIGH-EFFECTIVENESS RECUPERATORS FOR WASTE HEAT RECOVERY

By
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California, a division of The Garrett Corporation,
and Roy Stake, The Garrett Corporation

(Presented at the Waste Heat Utilization Conference sponsored by the Engineering Foundation Conference at New England College, in Henniker, New Hampshire on August 13-18, 1978)

INTRODUCTION

Direct heating processes using natural gas and fuel oil in the U.S. industrial sector represent 11.5 percent of the national consumption of energy. Of this amount, 2.7 quads per year (3.6 percent of the national consumption of energy) are being dumped directly into the atmosphere at flue gas temperatures above 1000°F (1 quad/yr = 10^{15} Btu/yr $\approx$ 500,000 Bbl/day). A new concept high-effectiveness, high-temperature stainless steel recuperator that can economically recover a significant amount of this heat has recently become available. This paper describes the new recuperator and its ability to recover much of this energy at very attractive payback periods.

In a recent survey conducted by AiResearch Manufacturing Company for the Department of Energy* it was concluded that high-temperature, high-effectiveness recuperators could save a minimum of 350 x 10^{12} Btu, or 0.35 quads, of energy per year. This amount of energy is worth, at $2.60 per million Btu, $910,000,000 annually. The cost of recuperators to conserve the energy, including associated ducts and installation hardware, would be approximately $530,000,000.

The amount of stainless steel required to fabricate the heat exchangers is not excessive: approximately 35,000 tons of stainless steel would be required. This amount represents less than 4 percent of one year's production of stainless steel in the U.S.

HIGH-TEMPERATURE WASTE HEAT RECUPERATION

Table 1 lists by Standard Industry Classification (SIC) the energy saving potential of selected industries using 85 percent effective recuperation for combustion air preheat. Steel, nonferrous metals, glass, and cement industries represent the major potential savings. Figure 1 shows the distribution of fuel consumption of these industries by flue gas temperature range. The two lower flue gas temperature ranges accounted for nearly half of the total amount of oil and gas used. Some of the processes involved at these temperatures were in preheating and open-hearth steel making with regeneration, drying, enameling, baking, and some heat treating. The medium-temperature processes included

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*"Survey of Potential Energy Savings Using High Effectiveness Recuperators for Waste Heat Recovery from Industrial Flue Gases," AiResearch Report 77-14483, prepared for Department of Energy under Contract EC-77-C-07-1642, October 15, 1977.

TABLE 1

ENERGY SAVING POTENTIAL BY 3-DIGIT SIC USING 85 PERCENT EFFECTIVENESS RECUPERATORS FOR COMBUSTION AIR PREHEAT

SIC	Industry Group	Gas and Oil Consumed, Btu x 10^{12}	Potential Savings with Recuperation, Btu x 10^{12}	Percent Savings
321	Flat glass	50.77	9.39	18.5
322	Pressed or blown glass	183.01	19.19	10.5
323	Products of glass	7.71	2.39	31.0
324	Cement, hydraulic	249.94	13.46	5.3
325	Structural clay products	102.09	0.54	0.5
326	Pottery and related products	17.93	0.70	3.9
331	Blast furnace, basic steel products	910.41	195.18	21.4
332	Iron and steel foundries	76.79	7.17	9.3
333	Primary nonferrous metals	265.66	40.07	15.1
334	Secondary nonferrous metals	26.00	6.89	26.5
335	Nonferrous rolling and drawing	110.11	26.39	24.0
336	Nonferrous foundries	28.05	8.40	29.9
339	Misc. primary metal products	19.06	2.52	13.2
341	Metal cans, shipping containers	27.84	0	0
343	Plumbing, heating except elec.	8.23	0.73	8.9
344	Fabricated struc. metal products	50.79	0.53	1.0
345	Screw machine products, bolts, etc.	12.29	0	0
346	Metal forgings and stampings	64.02	13.89	21.7
347	Metal services, NEC	19.36	0	0
349	Misc. fabricated metal products	28.86	0.67	2.3
358	Refrigeration and service machine	22.97	0	0
363	Household appliances	21.86	0.86	3.9
372	Aircraft and parts	28.88	1.01	4.6
	TOTALS	2,332.60	349.98	15.0

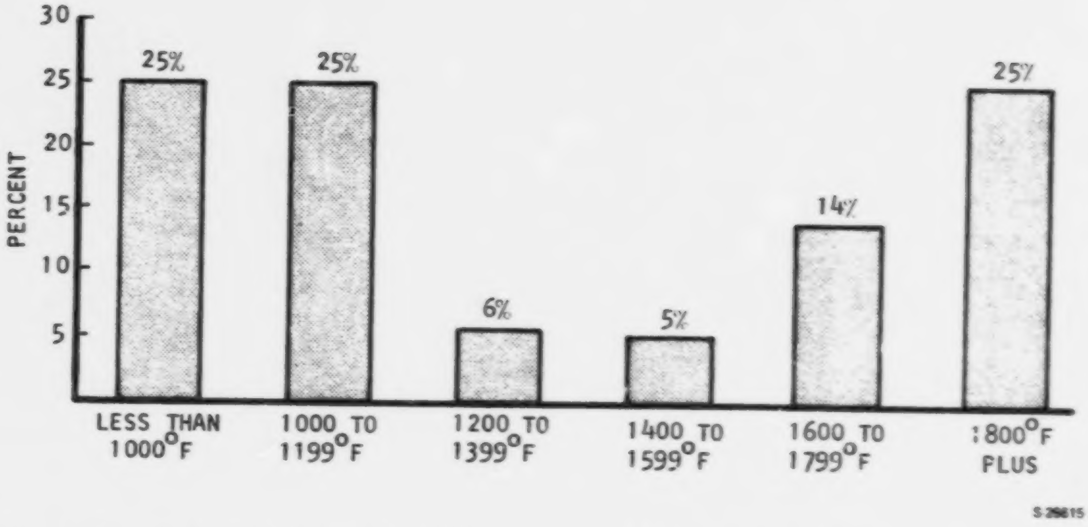


Figure 1. Distribution of Fuel Consumption by Flue Gas Temperature Range

heat treating, annealing, hardening, billet reheat, drying, and forging. In the high-temperature range (1800°F and above), the major sources of energy consumption were smelting and melting of metal and metal reheating. The industries using these processes (primary and secondary metals) have the greatest potential for fuel savings using recuperation. Over 200 x 10^{12} Btu/year may be saved from this high-temperature range alone.

Stainless steel recuperators are limited to metal temperatures of about 1500°F for long life. This limits the maximum flue gas temperature to 1550°F for high-effectiveness units. The two higher temperature ranges are above 1550°F; however, stainless steel recuperators can be used, with some thermodynamic penalty, by diluting the hot gas with ambient temperature air to achieve a mix temperature of 1550°F.

In 1977 AiResearch Manufacturing Company introduced a new concept in high-effectiveness recuperators. The specific production application of the technology was for the recuperation of large industrial gas turbines. The technology has been applied to flue gas recuperators and is being demonstrated on an aluminum remelt furnace with a flue gas temperature ranging between 1300° and 1800°F.

GAS TURBINE RECUPERATORS

In a recuperated gas turbine cycle (Figure 2), the recuperator functions to transfer heat from the exhaust gas to the air after the air leaves the compressor and before it enters the burners. The curves in Figure 3 show the improvement in efficiency that results from recuperating the cycle. Recuperated turbines are usually in the pressure ratio range of 6:1 to 10:1, where

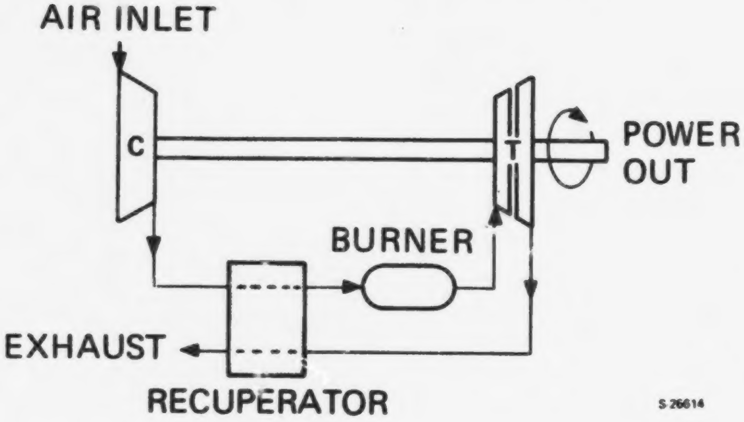

AIR INLET
C
T
POWER OUT
BURNER
EXHAUST
RECUPERATOR
S 26614

Figure 2. Gas Flow Diagram of a Recuperated Cycle Two-Shaft Gas Turbine

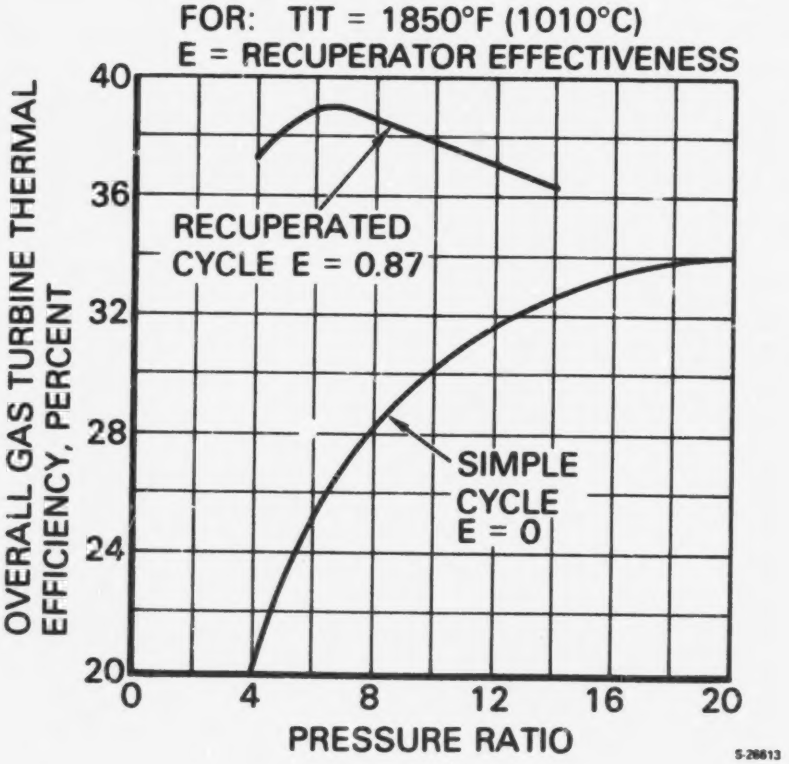


Figure 3. Typical Improved Efficiencies from Recuperating the Cycle

the improvement in cycle efficiency due to recuperation ranges from 20 to 50 percent. Also note from Figure 3 that a low pressure ratio, recuperative cycle turbine has a higher efficiency than a high pressure ratio, simple cycle turbine. In addition to higher thermal efficiency, low pressure ratio turbines and compressors are less complex both mechanically and aerodynamically.

The basic type of construction is shown in Figure 4. It is a brazed, plate-fin, counterflow, integral manifold, formed tubeplate heat exchanger (U.S. Patent No. 4,073,340). Counterflow construction is achieved by plain turning fins in both the air and gas passages to turn and distribute the gas and air into the counterflow section. The high effectiveness is achieved by the high-performance lanced-and-offset fins in the counterflow section of both the air and gas passes. Integral manifold construction is accomplished by extensions of the tubesheet, which forms complete air manifolds when the passages are brazed together. For high-pressure applications such as gas turbine recuperators, the integral manifold concept is structurally superior to manifolds which are welded in the cores. Formed tubeplate construction means that grooves are formed into the tubeplates, resulting in complete separation of the air and gas passages when stacked and brazed. Formed tubeplate construction is both more economical and more durable than the "log-cabin" construction in which solid bars are laid between tubeplates to separate the two fluid passages.

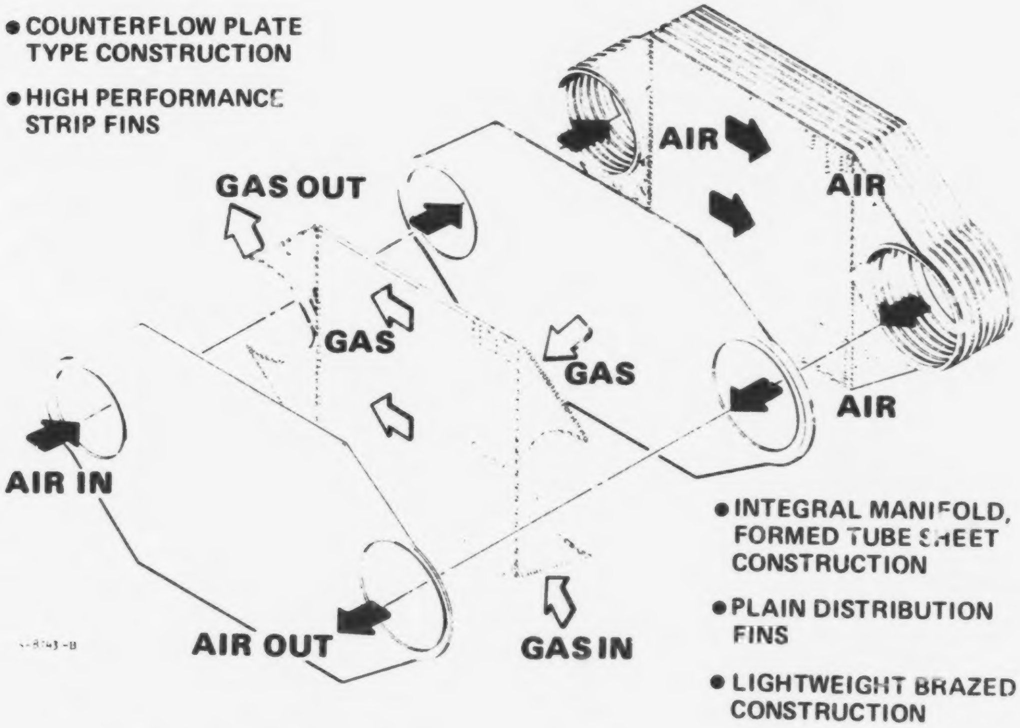

• COUNTERFLOW PLATE TYPE CONSTRUCTION
• HIGH PERFORMANCE STRIP FINS
GAS OUT
AIR
AIR
GAS
GAS
AIR
AIR IN
• INTEGRAL MANIFOLD, FORMED TUBE SHEET CONSTRUCTION
• PLAIN DISTRIBUTION FINS
• LIGHTWEIGHT BRAZED CONSTRUCTION
AIR OUT
GAS IN

Figure 4. Brazed, Plate-Fin, Counterflow, Integral Manifold Formed Tubeplate Heat Exchanger

During fabrication of the assembly, a high-temperature nickel-base braze alloy is placed on both sides of the tubeplates. The stack of tubeplates and fins is then placed in a high-temperature furnace. All brazed joints, including the fins to the plates, are made simultaneously by the furnace brazing operation.

Photographs of both the air and gas passages are shown in Figure 5. Each brazed core contains 33 gas passages and 32 air passages, and is capable of providing 85 percent effectiveness for air and gas flow rates of 10 pps per core.

F-26189
GAS SIDE
AIR SIDE

Figure 5. Regenerator Core Passages

Figure 6 shows the type of fins used in the counterflow section; this is the highest performing type of fin available. This same type of fin, except smaller in size, is used in AiResearch aerospace applications where performance is critical. A new boundary layer is started at each offset, resulting in heat transfer coefficients that are more than twice as large as with plain fins. For industrial applications, the benefits of higher heat transfer coefficients are reduced size and more economical use of materials.

Figure 7 shows a brazed recuperator assembly. Insert A is a section at the core manifold showing the formed tubesheet construction. Note compatible gage thicknesses are used throughout to avoid thick-to-thin thermal fatigue problems. Insert B of Figure 7 is a core sample showing construction of the counterflow section of the core. Efficient heat transfer is accomplished by the brazed metallurgical bond between the fins and the plates. For applications in which high pressure is present on one side, such as gas turbine recuperators, the fin acts as a structural tie between the plates to offset the pressure loading. For applications in which both fluid streams are near atmospheric pressure, both fins would be similar to the large gas fin in Figure 7.

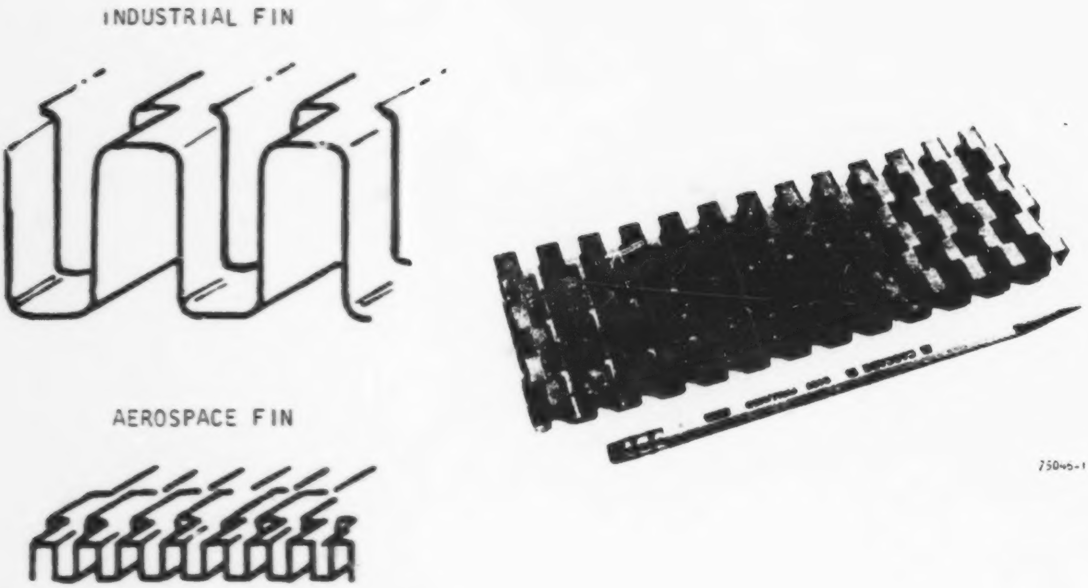
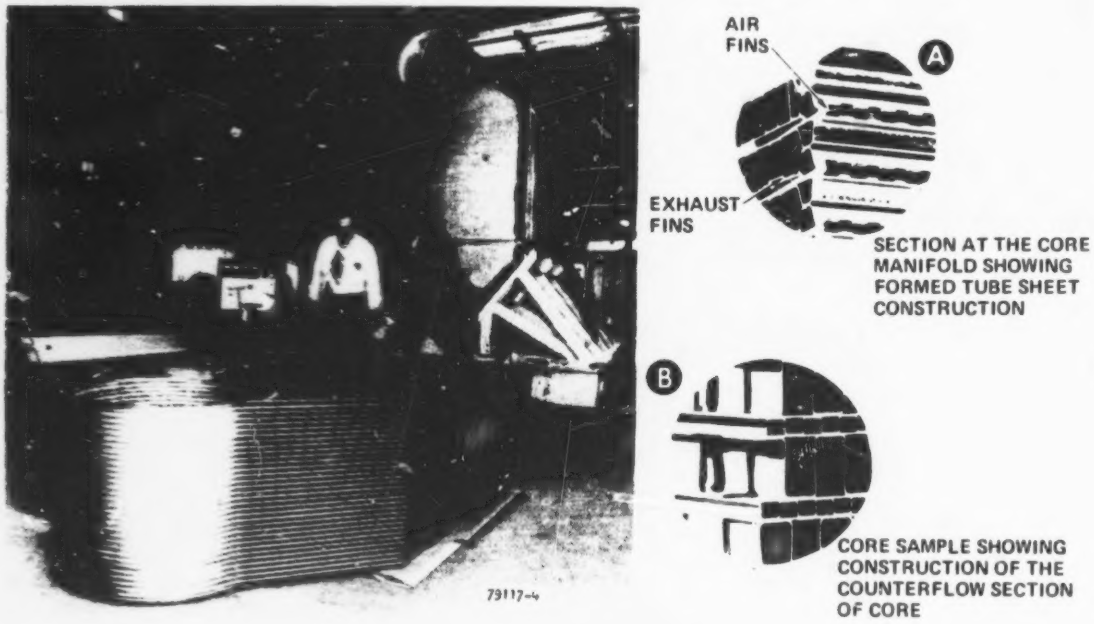

INDUSTRIAL FIN
AEROSPACE FIN
75046-1

Figure 6. Fins for Counterflow Section


AIR FINS
A
EXHAUST FINS
SECTION AT THE CORE MANIFOLD SHOWING FORMED TUBE SHEET CONSTRUCTION
B
CORE SAMPLE SHOWING CONSTRUCTION OF THE COUNTERFLOW SECTION OF CORE
79117-4

F-27317

Figure 7. Brazed Recuperator Core Assembly

Figure 8 shows an assembly of 12 cores in a recuperator designed for 120 pps air and gas flow. The cores are welded together into a 6-core assembly, which is mounted in a low carbon steel case. The case is insulated on the inside, and expansion joints (bellows) are used to connect the core to the pipe flanges. The core is hung on flexible straps; these straps, along with the expansion joints, allow the core to expand and contract without restraint during temperature changes that occur during startup and shutdown.

Figure 9 is a photograph of a six-core assembly designed for gas flow vertically upward through the cores. Figure 10 is a photograph showing the six-core module in its case. The case is insulated on the inside, and the forward structure is strong enough to transfer the large mechanical loads from the 20- and 24-in. high-pressure air pipes directly to the foundation. Manways (not shown) are provided on the opposite end of the assembly to provide access for field inspection and field repair, if necessary.

Figure 11 shows two halves being installed on a gas pipeline pumping station, and Figure 12 shows the completed installation.

Figure 13 shows a variety of installation concepts for a gas turbine recuperator for air and gas flows of 50 pps to 250 pps. Other installation concepts can be accommodated for other types of waste heat recovery applications.

PROJECT DESCRIPTION--DOE FLUE GAS RECUPERATOR DEMONSTRATION PROGRAM

In early 1977 the Department of Energy (then ERDA) awarded a contract (EC-77-C-07-1642) to the AiResearch Heat Transfer Systems product line to fabricate and install a high-effectiveness recuperator for combustion air preheat in an aluminum remelt furnace and demonstrate the fuel savings available.

Figure 14 illustrates the recuperator installation schematically. Foundry air is drawn in through a filter by a blower and supplied to the counterflow plate-fin recuperator in the furnace stack. In the recuperator, the combustion air is heated, cooling the flue gas. The heated air is then ducted to the burner, which is fired by either natural gas or fuel oil. The flame temperature, which normally rises by the use of preheated air, is maintained at near stoichiometric temperature by the use of some excess air, which will ensure complete combustion. The flue gas temperature is controlled by stack dilution to approximately 1550°F, using a motor operated control valve and stack temperature sensor.

The 1550°F stack gas temperature allows the demonstration of a high-effectiveness recuperator without recuperator metal temperature or combustion air preheat temperature being out of the range of present materials technology. Current preheat temperature is about 1450°F.

The heated air enters into combustion with the supplied fuel, transferring heat to the molten aluminum by radiation and convection. The flue gas exits up the stack, is diluted to 1550°F, and passes through the recuperator, transferring the heat to the air on the cold side of the recuperator. An exhaust fan at the top of the stack makes up for the pressure drop in the recuperator so that the furnace is maintained at a near neutral pressure.

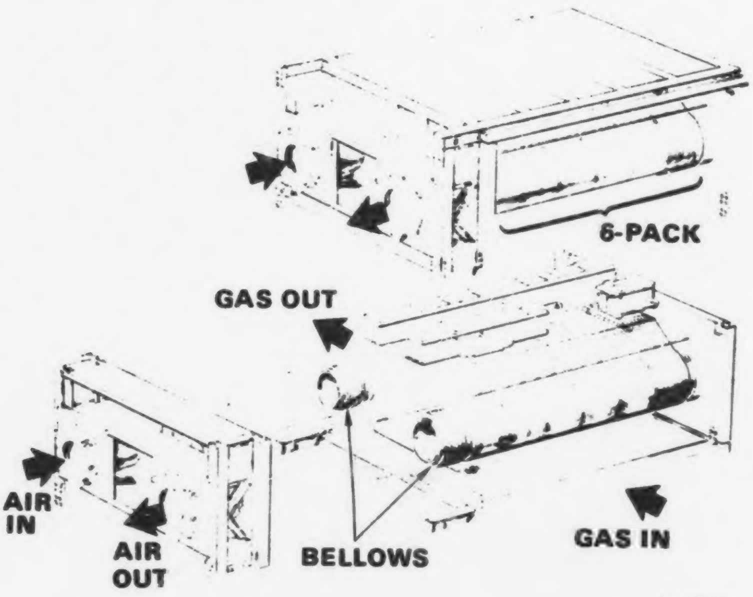

6-PACK
GAS OUT
AIR IN
AIR OUT
BELLOWS
GAS IN
F-26222

Figure 8. Recuperator with Two Six-Core Assemblies

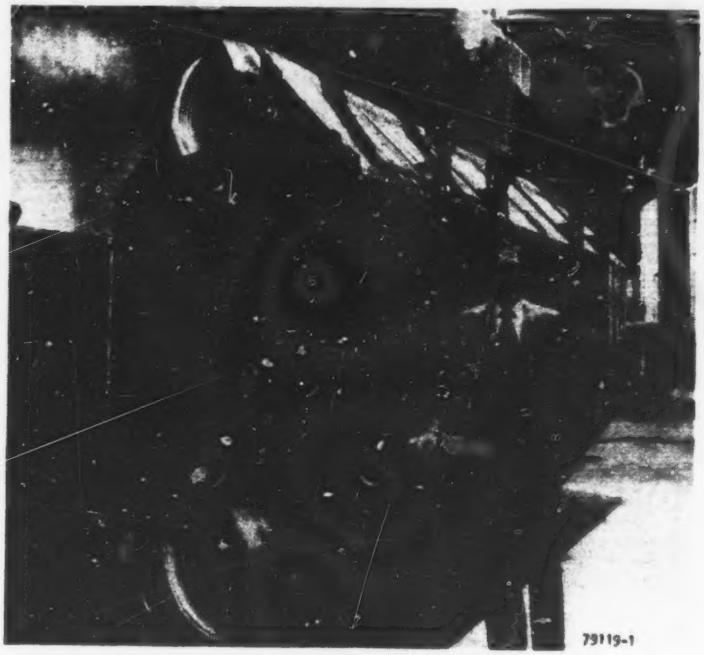


Figure 9. Regenerator Unitized Assembly

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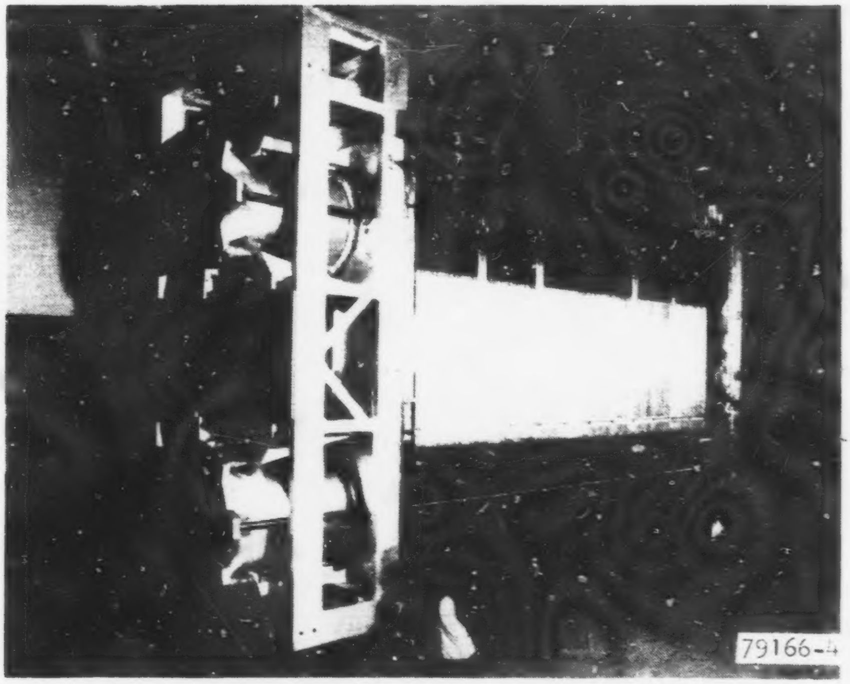

Figure 10. One-Half Recuperator Module

F-27315

Figure 11. Installation of Regenerator

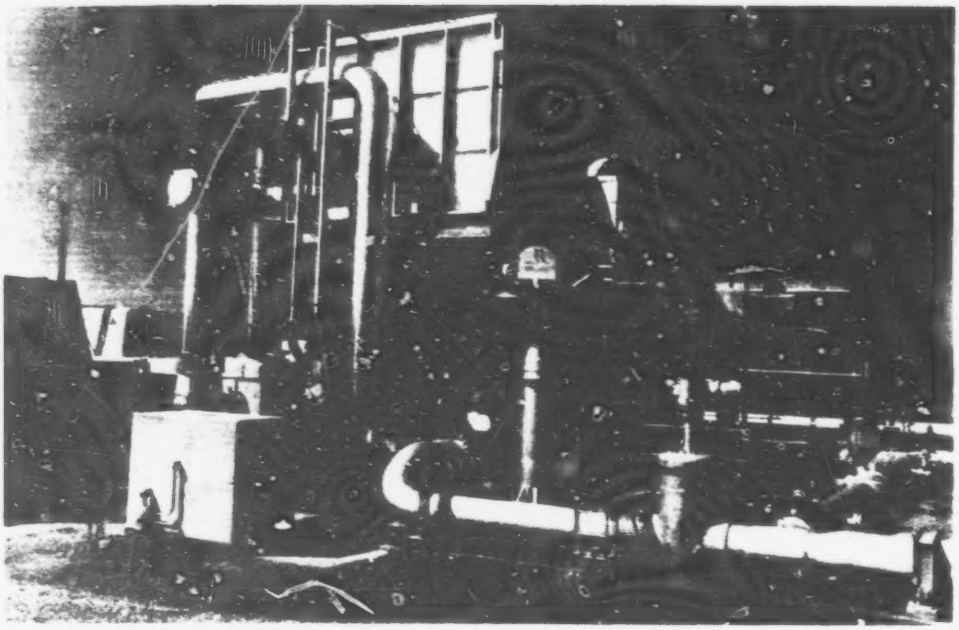

Figure 12. Installed 15,000-hp AiResearch Regenerator

A VARIETY OF INSTALLATIONS AND CAPABILITIES ARE ACCOMMODATED BY THE GARRETT MODULAR REGENERATOR CONCEPT
ONE HALF MODULE VERTICAL EXHAUST GAS FLOW
ONE HALF MODULE HORIZONTAL EXHAUST GAS FLOW
BASIC MODULE END TO END HORIZONTAL EXHAUST GAS FLOW
BASIC MODULE END TO END VERTICAL EXHAUST GAS FLOW
BASIC MODULE HORIZONTAL EXHAUST FLOW
TWO MODULES HORIZONTAL EXHAUST GAS FLOW
BASIC MODULE OVERHEAD EXHAUST FLOW

Figure 13. A Variety of Installed Concepts of the Basic Recuperator Module

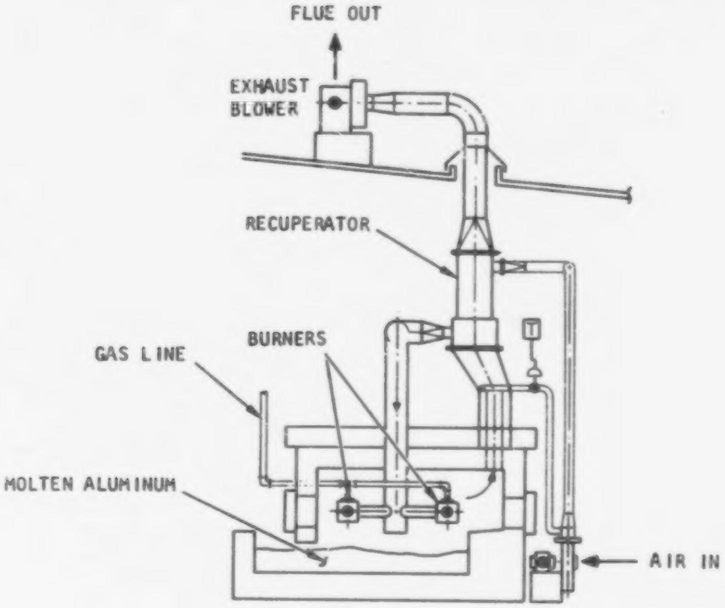

Figure 14. High-Effectiveness Recuperator Demonstration

During the demonstration period, recuperator inlet and outlet temperatures of the flue gases and combustion air are being monitored. Pressure drop data are taken across both sides of the recuperator. Fuel flow also is monitored.

The demonstration of the recuperator has proceeded as planned. The recuperator was installed at the beginning of January 1978; Figure 15 illustrates the recuperator installation. The furnace was put on-line on January 13, 1978. Since that time the recuperated furnace has been operated entirely by production personnel, with no change in procedure from unrecuperated operation. Fuel consumption rate at high fire has been reduced from 3.0 million Btu per hour to less than 2.0 million Btu per hour, and the rate of aluminum melting has been maintained the same or increased. The overall fuel savings has been 30 percent. Figure 16 illustrates the specific fuel consumption of the furnace which has been measured before and after recuperation. Measured heat transfer effectiveness of the recuperator is over 85 percent.

The recuperator, which has been fabricated, installed, and tested during the performance of this contract, is of the plate-fin type, with a significant amount of secondary surface enhanced by the use of offset fins of optimum configuration for the gas flows and pressure drops available. This construction has been developed by AiResearch for the aircraft and gas turbine industry where low weight and a compact configuration are required. A photograph of the heat exchanger core is shown in Figure 17. A photograph of the instrument panel of the instrumentation console used to monitor the test is shown in Figure 18.

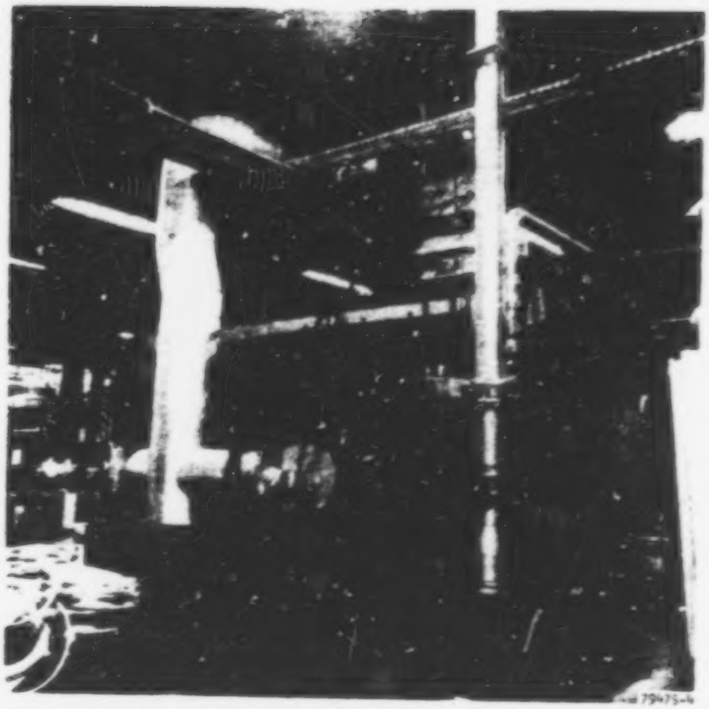

Figure 15. Recuperator Installation on McGowan Furnace

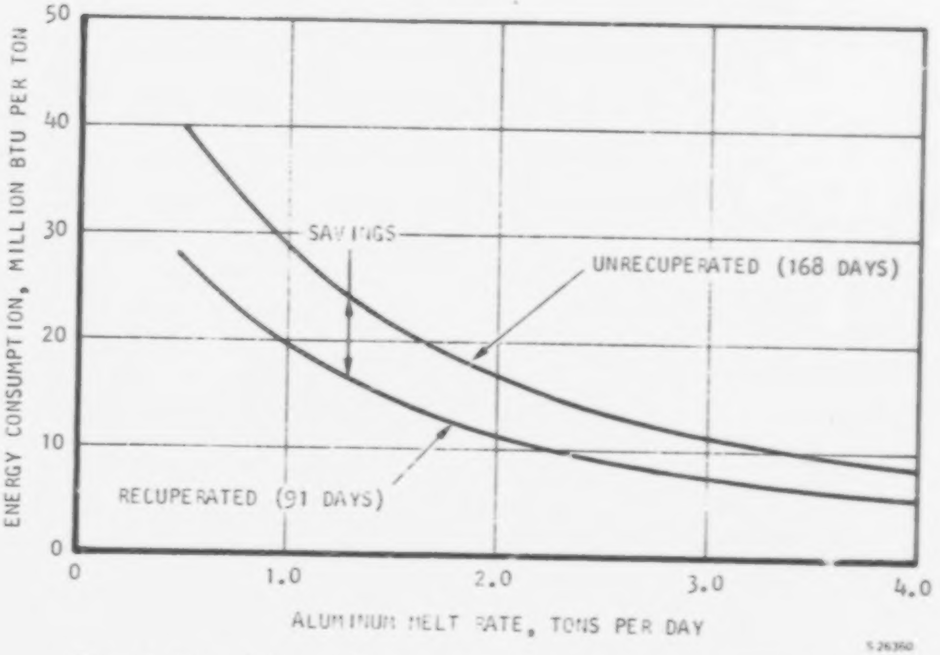

Figure 16. McGowan Furnace Specific Fuel Consumption

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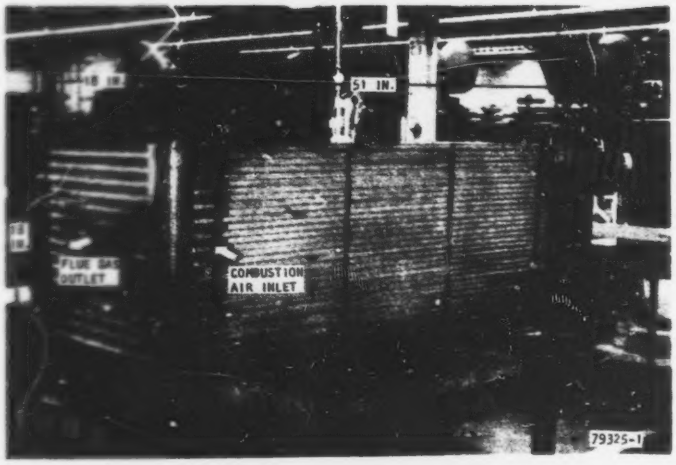
51 IN.
FLUE GAS OUTLET
COMBUSTION AIR INLET
79325-1

Figure 17. Completed Recuperator Core

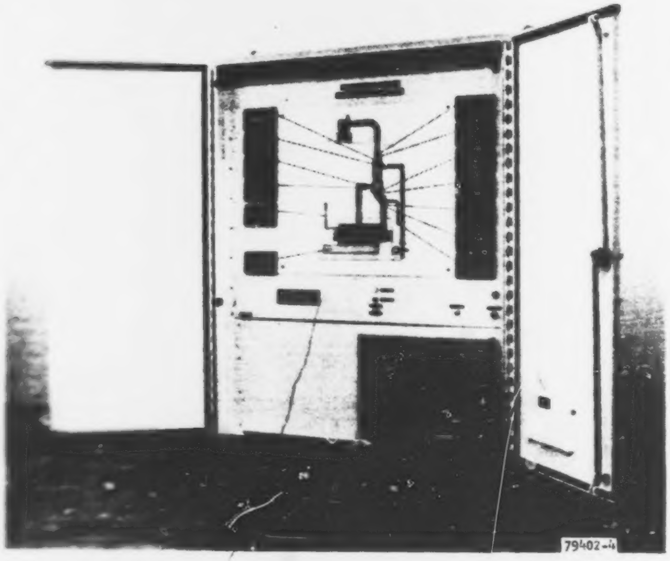

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Figure 18. Instrumentation Panel

The demonstration program will continue during 1978. Data will continue to be taken and analyzed for comparison with the documented performance of the unrecuperated furnace. To date, no change in recuperator performance has been observed and the measured savings have been as predicted.

CONCLUSIONS

The dramatic increase in cost and scarcity of hydrocarbon fuels has reemphasized the need for a high-effectiveness recuperator for waste heat recovery. Present furnace technology is the product of a lengthy production-oriented low-fuel-cost era during which there have been either no heat recovery systems or, at best, systems of less than maximum efficiency. A new technology is available in the form of a high-effectiveness compact stainless steel recuperator to greatly increase the recovery of waste heat.

Plate-fin heat exchangers such as the demonstration unit have reached a very high level of sophistication as a result of requirements in the aircraft and gas turbine industry for compact, lightweight, low-cost units. Heat transfer effectiveness in the range of 80 to 90 percent is readily obtainable, and the volume of these high-effectiveness units is less than recuperators currently used for the same application. Because of the reduced volume, installation is often possible where low-effectiveness crossflow or radiant tube units will not fit. Most important, the high-effectiveness recuperators increase fuel savings significantly, making their widespread application important for energy conservation.

It is estimated that 350 x 10^{12} Btu per year of energy (valued at over $900,000,000 per year) can be saved by the use of $530,000,000 worth of high-effectiveness recuperators. This savings is sufficient to supply the fuel needs of 5 million typical U.S. homes.

ACKNOWLEDGEMENTS

The authors wish to thank Jim Lohay, Project Engineer at AiResearch Manufacturing Company, for his assistance and counsel in preparing this paper.

PATENTS

The recuperators discussed herein are the subject of U.S. patent No. 4,073,340 and corresponding foreign patents. Additional U.S. and foreign patent applications are pending.

[redacted]

DOE/MTI COOPERATIVE VENTURE TO DEVELOP A PRODUCT FAMILY FOR INDUSTRIAL WASTE HEAT POWER RECOVERY

by

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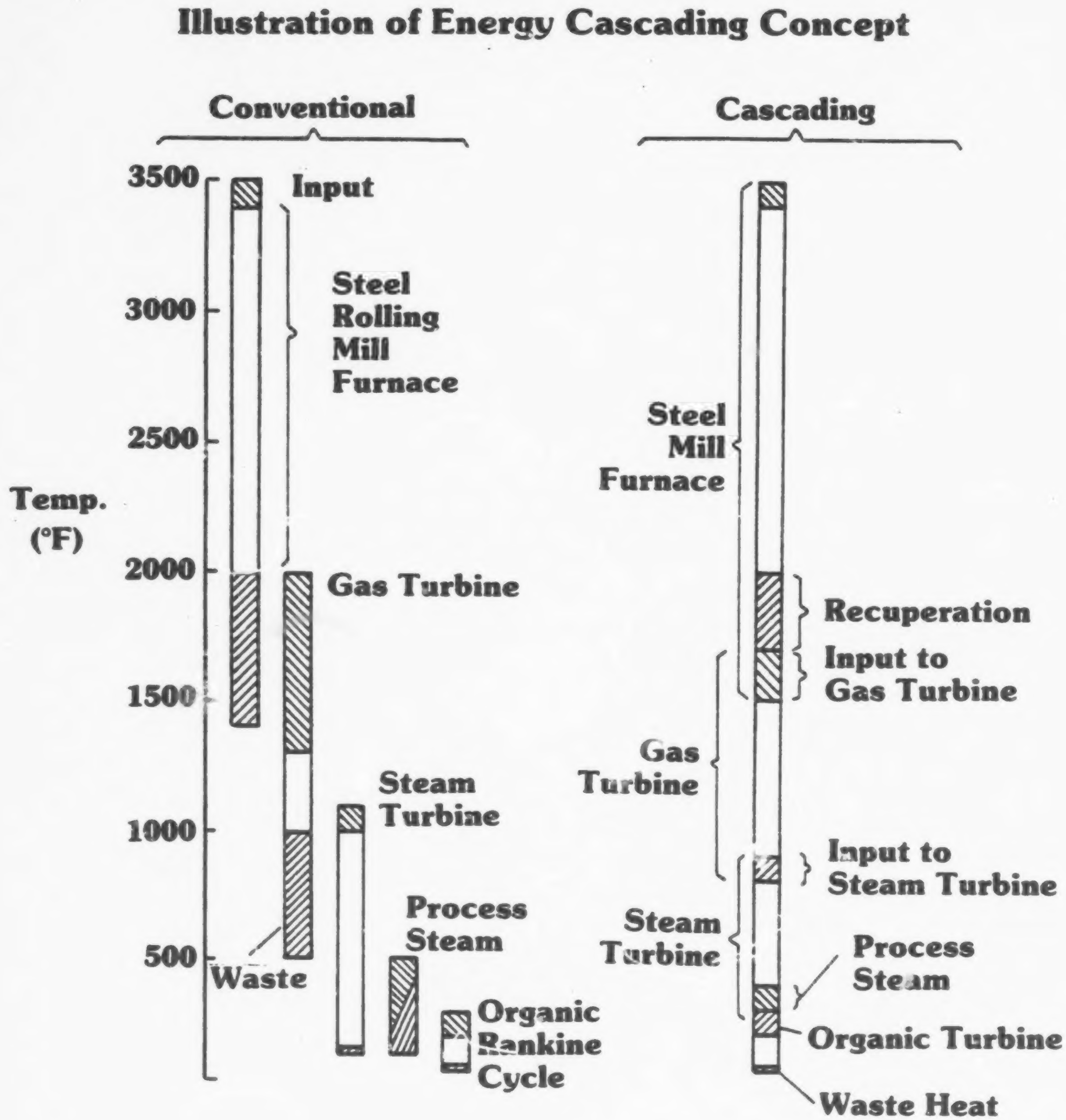

Illustration of Energy Cascading Concept

Conventional
Cascading
3500
3000
2500
2000
1500
1000
500
Temp. (°F)
Input
Steel Rolling Mill Furnace
Gas Turbine
Steam Turbine
Process Steam
Waste
Organic Rankine Cycle
Steel Mill Furnace
Recuperation
Input to Gas Turbine
Gas Turbine
Input to Steam Turbine
Steam Turbine
Process Steam
Organic Turbine
Waste Heat

MTI MECHANICAL TECHNOLOGY INCORPORATED

Figure 1

Example of Partial Energy Cascaded Techniques

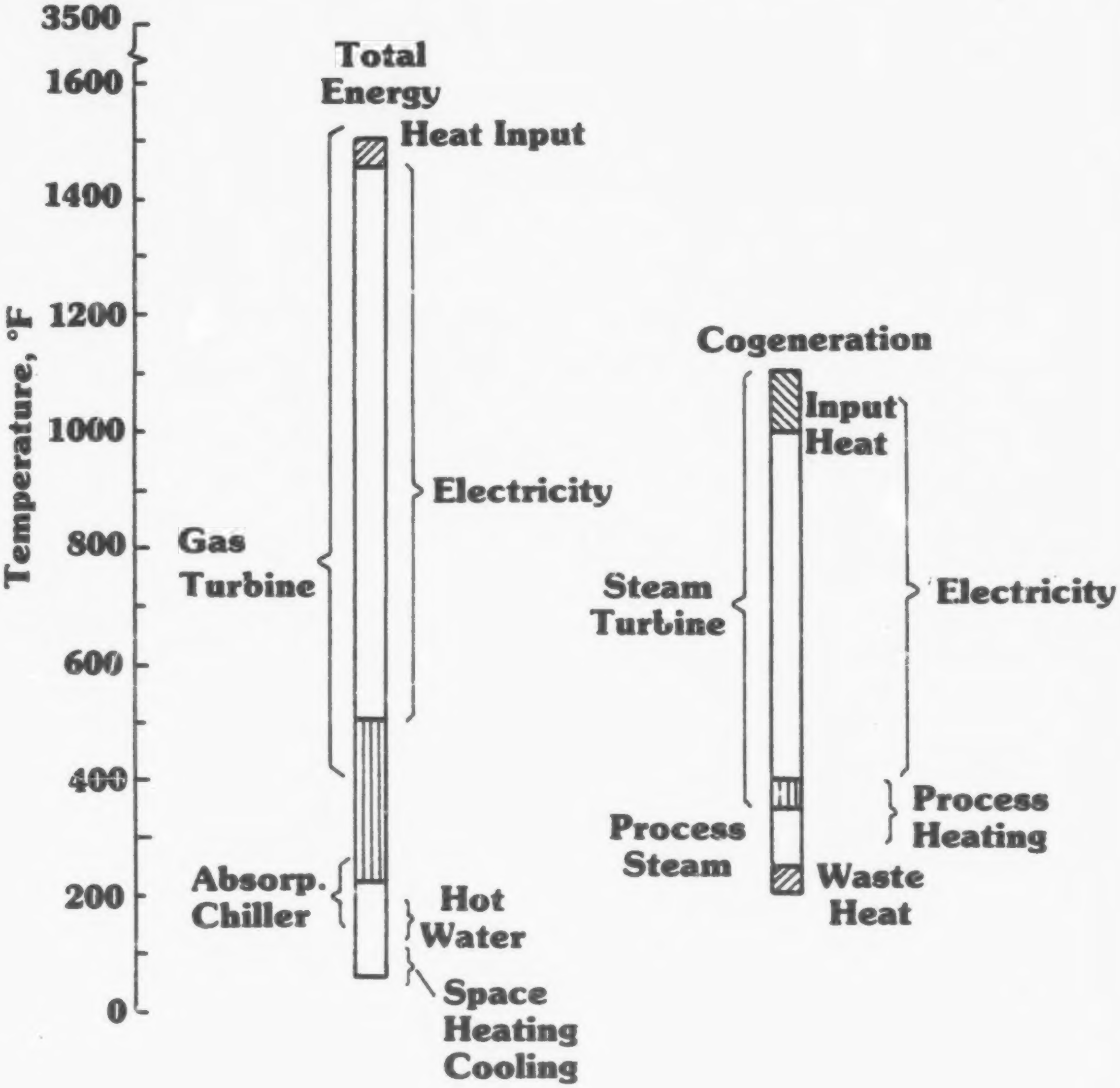

Figure 2

1.0 INTRODUCTION

Large amounts of energy are wasted each year when the chemical energy in fuels is transformed into thermal energy for heating materials or homes or for the production of electrical and mechanical power. This waste occurs because the energy potential in the fuels is not properly matched to the need. Examples of this are the use of a 3500°F flame to heat a home to 68°F or the use of that 3500°F flame to produce 300°F steam. More easily recognized examples of this waste are the 700°F exhaust of gas turbines and large stationary Diesel engines and vast amounts of 200°F to 350°F heat that are lost in chemical plants and refineries when the products of distillation are condensed. The efficient use of the chemical potential of fuels involves some type of energy cascading.

Energy cascading is the matching of the quality (temperature) of available energy to the needs of the tasks at hand. The available energy can be either direct fuel energy or rejected, and normally wasted energy, from another task. Actually, energy cascading is a restatement of good engineering design practice based on the Second Law of Thermodynamics. Conventional engineering design practice only considers the quantity of heat required for a task, not its quality. While we are used to seeing high system efficiencies expressed on the basis of conventional engineering practices, most processes are very inefficient when measured against energy cascading. A good example of this is a residential furnace. Under conventional design practice, the gas furnace is 75 percent efficient; under energy cascading, the gas furnace is only 5 percent efficient.

For a clearer illustration of the energy cascading concept, refer to Figure 1. On the left side of Figure 1 are listed five processes or cycles. Each process uses heat at a higher temperature and rejects heat at a lower temperature. The maximum quality or temperature of heat that occurs is that of a typical flame temperature of about 3500°F. While the steel rolling mill needs the full temperature available, the process temperature exhaust is about 2000°F. With recuperation, part of the energy can be used to pre-heat combustion air. However, this has not been general practice since energy costs were very low. An industrial gas turbine operates between a peak temperature of as high as 1800°F and exhausts as low as 500°F. Steam turbines operate over a range of about 1100°F to 140°F. Process steam is typically in the 300°F to 500°F range. Organic Rankine Cycles can operate from around 300°F down to the available cooling water temperature of 90°F or less. Each one of these five processes uses only part of the temperature range available. By combining all of the five processes so that the waste or exhaust of one is the input to the next, the full temperature-energy potential can be utilized.

Some of the current energy buzz-words of "total energy" and "co-generation" utilize only part of the total potential of energy cascading as shown in Figure 2. In total energy systems and co-generation applications, the first step (or top cycle) is the generation of electricity with the waste heat of the generation process being the input of the second step (or bottom cycle). In a total energy system, the bottom cycle is usually used for environmental conditioning (heating and cooling a building and heating water). In a co-generation application, the bottom cycle is usually process steam for an industrial plant. While combining the steps is more energy efficient than the separate steps, considerably more energy could be saved by fully applying the concept of energy cascading.

Utilization of waste heat for power recovery is part of the energy cascading concept. Waste heat power recovery applies Rankine systems to the rejected heat of the lowest task or cycle on the energy cascade. Currently, extremely large quantities of thermal energy that are discharged by industry are prime candidates for energy cascading with a Rankine Waste Heat Power Recovery System. The application of Rankine systems are only one of the numerous energy cascading approaches. There are also approaches to reduce energy consumption by changing various industrial processes. Diversity of approach is excellent from the national viewpoint and that of the industrial operator since there is a high probability that at least one technically and economically acceptable approach will be developed. However, from the viewpoint of a developer of specific alternatives, the large number of alternatives present a market risk in addition to technical risks.

As a developer of Rankine Power Recovery Systems, Mechanical Technology Incorporated (MTI) is extremely sensitive to the market risks. Therefore, MTI adopted the strategy of developing a family of Rankine Power Recovery Systems that can serve the entire range of waste energy streams. In the design of the product family, there will be an effort to maintain commonality of components and fabrication techniques to minimize product cost. Since it is obvious that only the most cost effective alternatives will be adopted, Rankine Systems must be produced and installed at an economically desirable cost to the end-user. Single unit, or single model, production of Rankine Power Recovery Systems may not result in sufficient manufacturing economies to make the Rankine Power Recovery Systems cost effective. Thus, in addition to minimizing the market risk, a product family may be the only way in which sufficient production volume can be attained to result in a cost effective system.

The Department of Energy (DOE) has the mission to develop and demonstrate energy conservation alternatives. The support of competing alternatives helps to ensure that there will be at least one technically acceptable alternative available and that the opportunity may exist to select the most cost effective alternative from among the technically acceptable concepts. A program to develop a family of Rankine Power Recovery Systems serves DOE's mission by providing another technical alternative to waste energy utilization over virtually the entire spectrum of available temperatures and discharge media as well as an alternative in the mid-temperature (400°F to 800°F) range to the other Rankine systems being supported by DOE.

This paper will briefly discuss the concept of the Rankine Power Recovery System, available waste streams as potential markets, user economics, product requirements, and DOE/MTI Cooperative Venture Program.

2.0 RANKINE SYSTEM CONCEPT

The Rankine Waste Heat Power Recovery System is basically a Rankine cycle engine. The overall system concept is shown in Figure 3. A waste heat stream is brought into the system in a heat exchanger which extracts heat from waste heat. This heat is transmitted to a Rankine turbomachinery conversion package which extracts some useful shaft power from the heat and rejects the rest of the heat to a lower temperature in a condensing heat exchanger. The amount of work that can be extracted is directly proportional to the efficiency of the cycle which is limited by the Second Law to the Carnot efficiency.

A more detailed presentation of the Rankine cycle and its features is presented in Figure 4. The Rankine cycle utilizes phase change of fluids such as water to approximate isothermal heat addition during the vaporization of water, (state points 2 to 3), and isothermal heat rejection during the condensing of steam, (state points 5 to 6). The variation of phase change properties of different fluids can be applied to achieve high efficiency over waste heat conditions ranging from high temperature steam (or even liquid metal) systems in central station utility plants, to a very low temperature ammonia system being considered for generating power from ocean thermal gradients. The conversion machinery in a Rankine system can be simple and reliable turbomachinery. The Rankine system uses external heat addition which makes it extremely adaptable to various types of heat sources whether it be fuel combustion, nuclear, solar, waste heat or geothermal. All of these features make the Rankine system intrinsically a very flexible system.

The Rankine system can be made even more flexible by cascading Rankine cycles with different fluids in the same system. This concept is termed the binary Rankine system and was developed by MTI for waste heat utilization under a DOE contract. A simplified schematic of the binary concept is shown in Figure 5. MTI's concept uses two commonly known fluids, water and R-11, to achieve a good and safe match between the Rankine cycle working fluid and the heat source and to achieve a rather simple, but high efficiency, turbomachinery component, as well as match the low condensing temperature available without high vacuums and large specific volumes. A more detailed schematic showing the actual

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Rankine Waste Heat Power Recovery System Concept

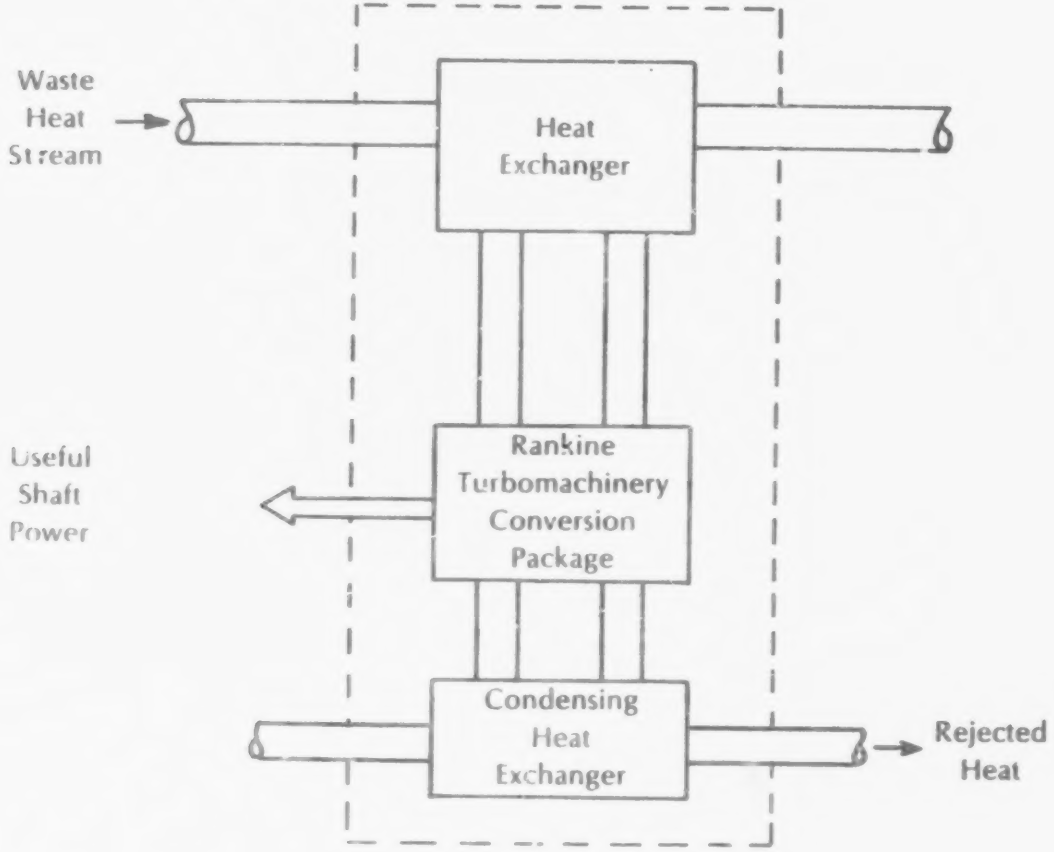
Waste Heat Stream
Heat Exchanger
Useful Shaft Power
Rankine Turbomachinery Conversion Package
Condensing Heat Exchanger
Rejected Heat

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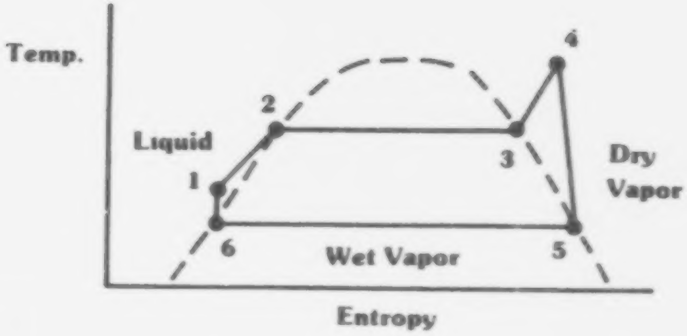
Figure 3

Rankine Cycle Features



- Two-Phase Cycle
- Many Potential Working Fluids
- Low Temperature System

- Simple System
- External Heat Addition
- Flexible

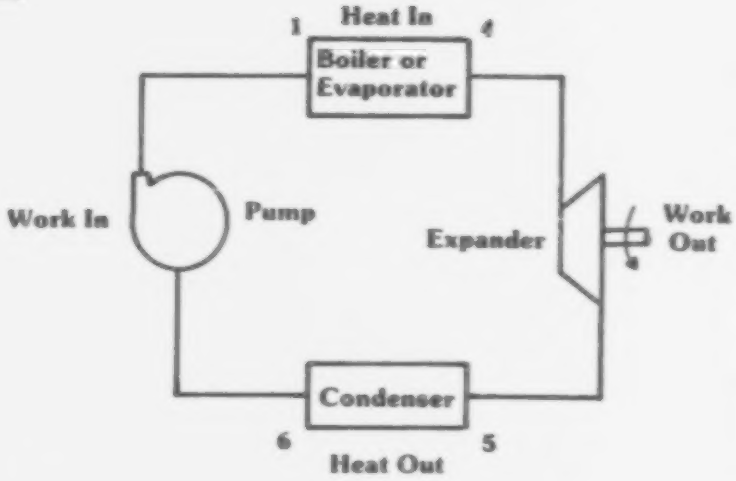

1
Heat In
4
Boiler or Evaporator
Work In
Pump
Expander
Work Out
Condenser
6
5
Heat Out

Figure 4

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Binary Cycle Simplified

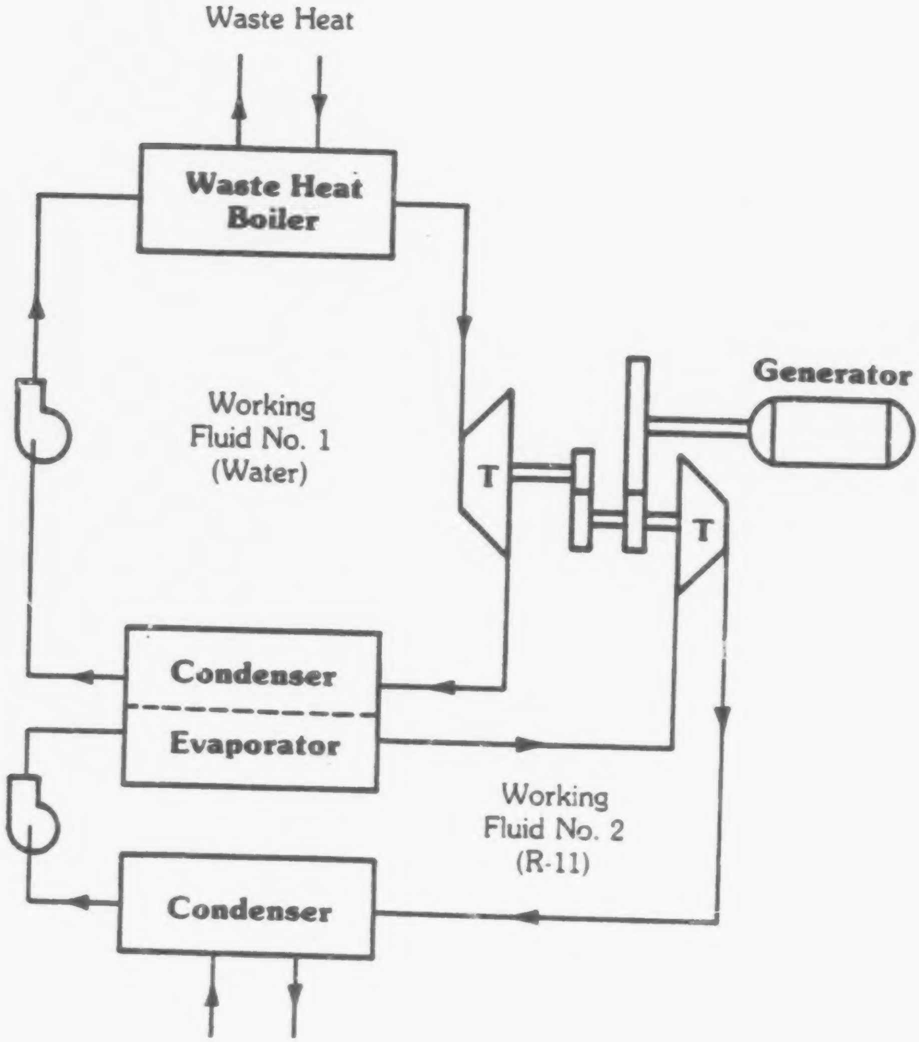
Waste Heat
Waste Heat Boiler
Working Fluid No. 1 (Water)
T
Generator
T
Condenser
Evaporator
Working Fluid No. 2 (R-11)
Condenser

Figure 5

state points and conditions of the unit developed by MTI is shown in Figure 6. It should be noted that all fluid pressures within the system are maintained above atmospheric pressure. This particular system has been built and successfully tested at MTI. The unit is currently being installed at the Municipal Power Plant of Rockville Centre, New York, (a municipality on Long Island, a few miles from J.F. Kennedy Airport) and is shown in sketch form on Figure 7.

Cycle Schematic for the MTI Binary Demonstration System

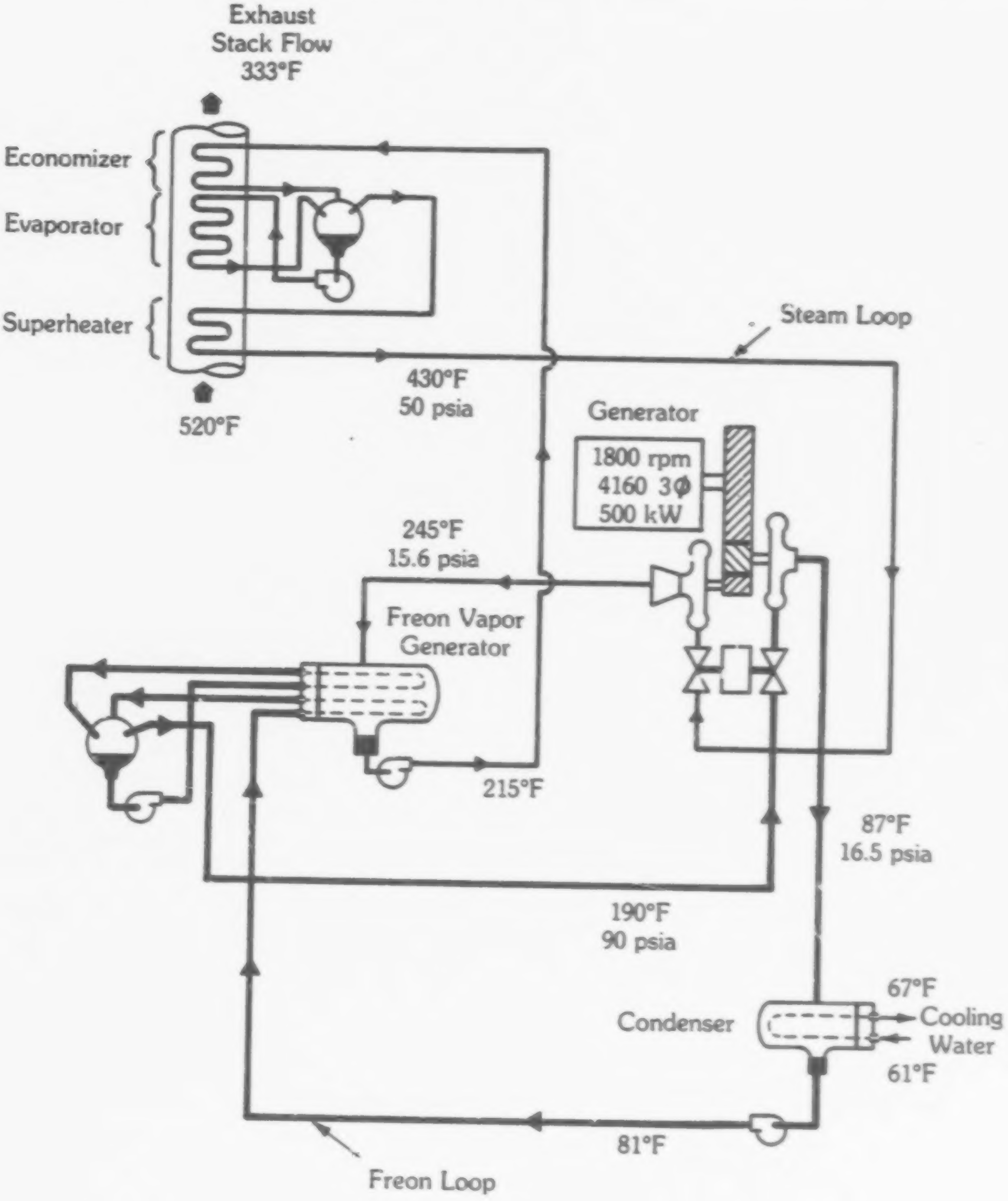
Exhaust
Stack Flow
333°F
Economizer
Evaporator
Superheater
520°F
430°F
50 psia
Steam Loop
Generator
1800 rpm
4160 3Φ
500 kW
245°F
15.6 psia
Freon Vapor
Generator
215°F
87°F
16.5 psia
190°F
90 psia
Condenser
67°F
Cooling
Water
61°F
81°F
Freon Loop

Figure 6

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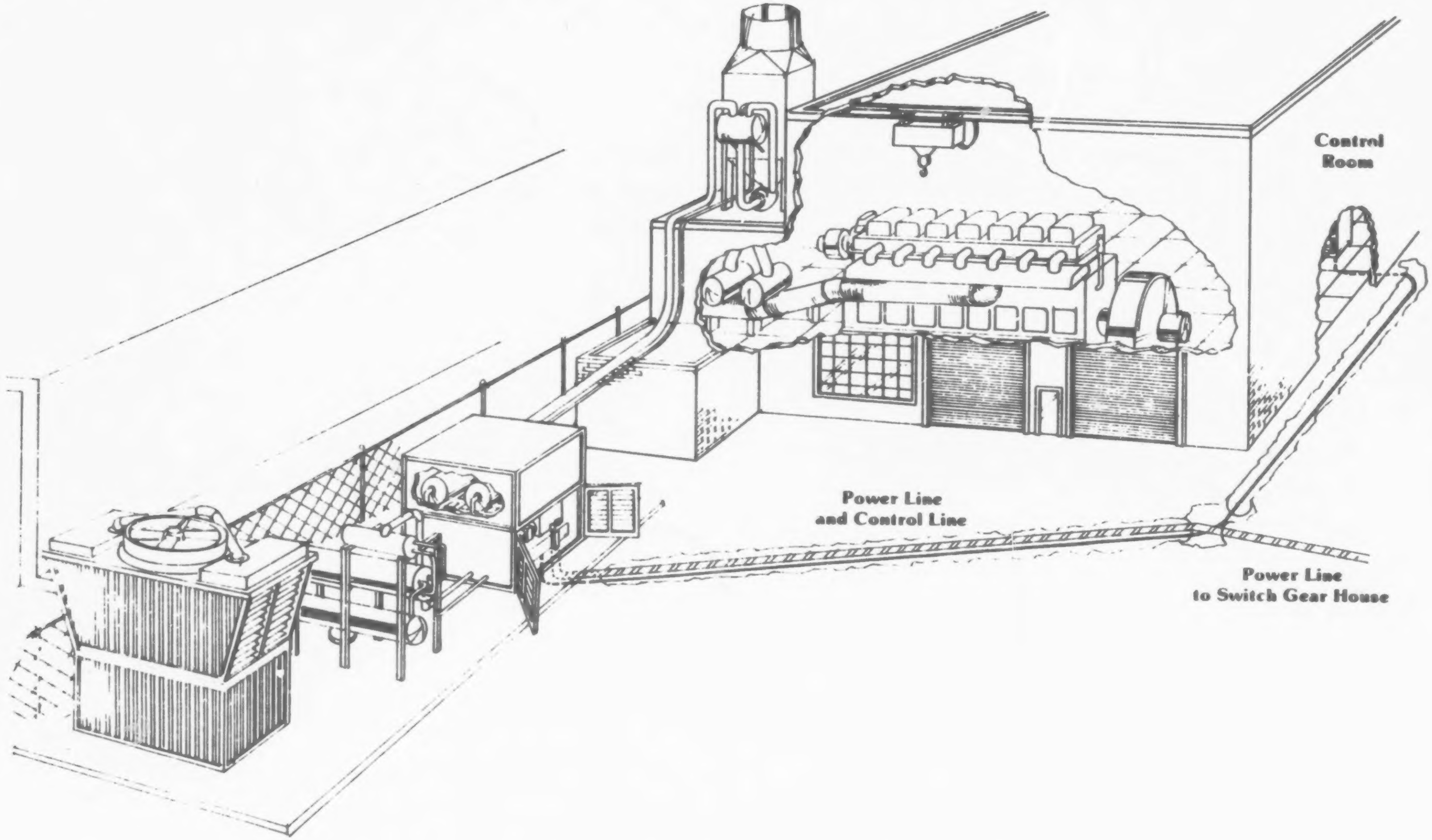
Control Room
Power Line and Control Line
Power Line to Switch Gear House

General Arrangement of MTI Binary Rankine Cycle System for Waste Heat Recovery/Electric Power Generator

Municipal Power Plant
Rockville Center, New York

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Figure 7

3.0 AVAILABLE WASTE STREAMS

The consideration of applying Rankine systems to waste heat utilization must include the characterization of waste heat flows. Fortunately, data is now becoming available regarding the quantity and temperature of industrial waste heat flows. Results of this DOE sponsored work is shown in Figure 8. Data in Figure 8 is presented as the amount of heat rejected per degree at a given temperature. This plot shows that about 70 percent of the heat rejected is between 100° to 600°F. Furthermore, there are concentrations of rejected heat, about certain obvious temperatures such as 100°F, for cooling water.

Thermodynamically, it is easier and more efficient to use high temperatures to produce power than low temperatures. Figure 9 presents the Carnot efficiency for various peak cycle temperatures with a 100°F condensing temperature. Shown in Figure 9, also, are the actual performance regimes of large Rankine and Brayton (gas turbine) engines and the expected performance of large Stirling engines. Examination of Figure 9 illustrates both the desireability of using high temperatures and that in the temperature range of most industrial waste heat streams, the Rankine cycle is the only cycle worth considering. If large Stirling engine technology was well developed, then it would be an ideal bottoming cycle for high temperature industrial waste streams.

The cost of Rankine systems varies with both available source temperature and total system capacity as shown in Figure 10. As the available temperature increases, the thermodynamic efficiency increases and more power is generated with the same physical machinery. Thus, the unit price (dollars per kw) decreases as the temperature increases. Similarly, the unit price of a Rankine system decreases as the total system capacity increases. This comes about since there is a certain amount of fixed cost for a system. Thus, the larger the total system the smaller the amount of fixed charges per unit output. This implies that large high temperature systems would have the lowest unit cost. In the electric utility industry this is well known and has been the key impetus for large central stations operating at maximum operationally feasible turbine inlet temperatures.

DISTRIBUTION OF INDUSTRIAL WASTE HEAT FLOWS

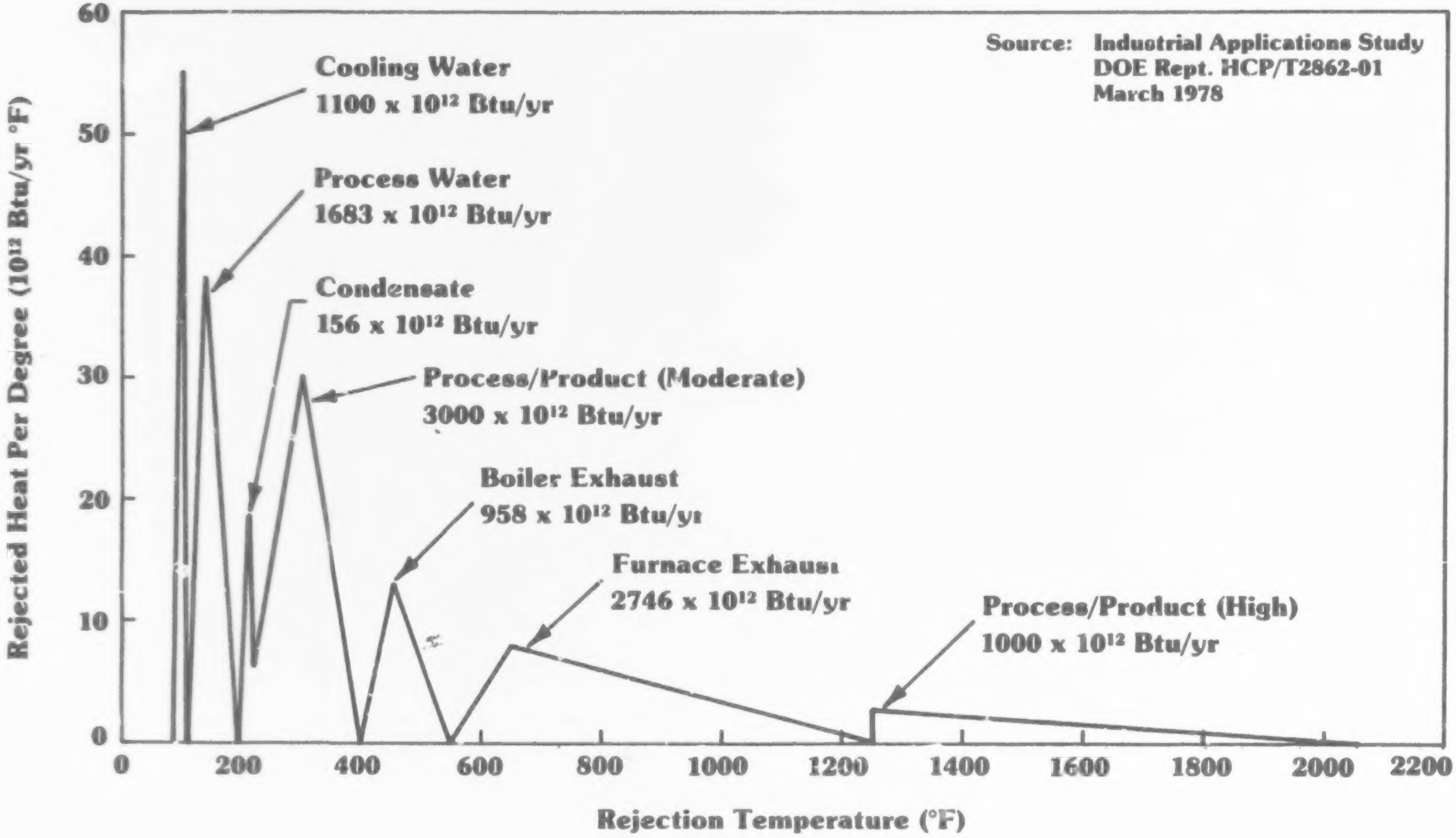

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Figure 8

THERMODYNAMIC POTENTIAL OF ENGINES

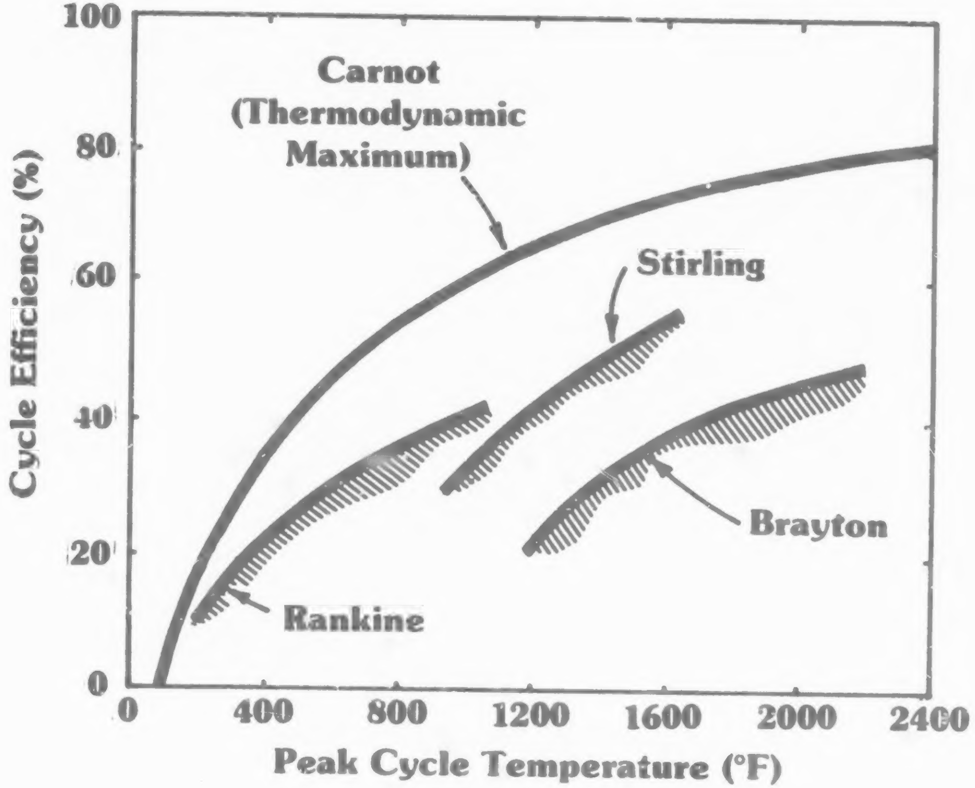

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Figure 9

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GENERAL DISTRIBUTION OF SYSTEM PRICE

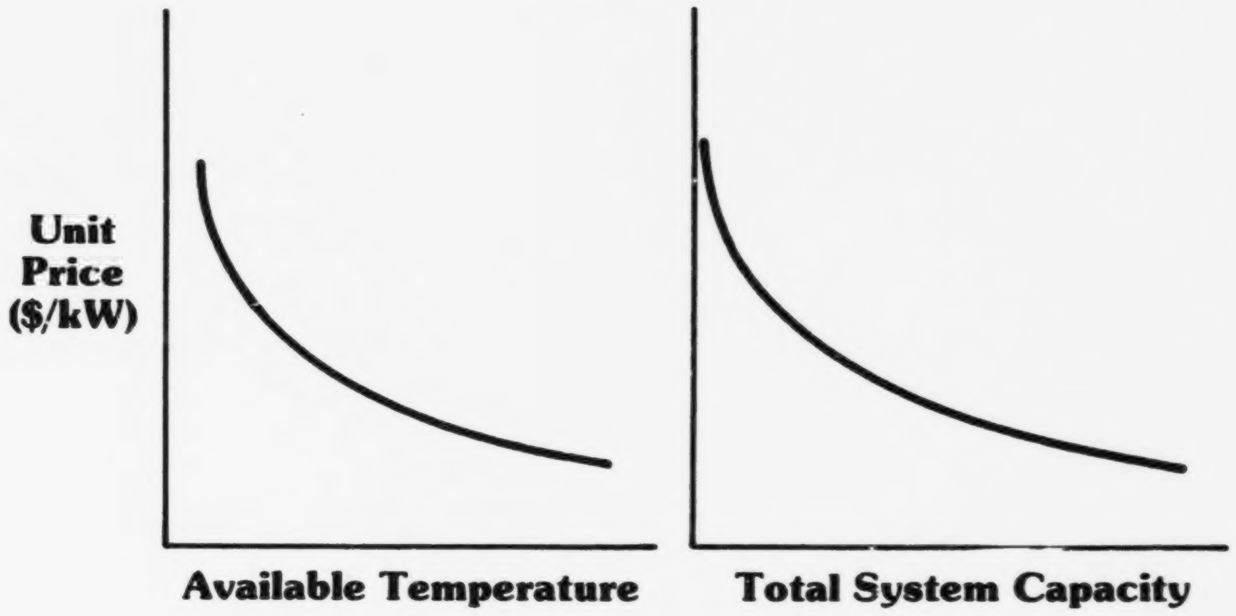

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Figure 10

Available waste heat flows, combined with the economics of thermodynamics and scale, produce a distribution of economically available waste heat flows for Rankine Power Recovery Systems shown qualitatively in Figure 11. The shape and mode of this economically available waste heat is a function of an individual company's market perspective, system performance and cost. This distribution, also, is not static.

Perception of the distribution of the waste heat flow plus product philosophy will result in a Rankine system product selection, illustrated in Figure 12. Product philosophy, in this case, is MTI's approach of using accepted working fluids without actual or perceived user risk. The selection, (or restriction), of working fluids to steam and the fluorocarbons define certain ranges for each unit as shown in Figure 12. For high temperatures, steam is clearly preferred; for low temperatures, fluorocarbons are clearly preferred. In the moderate temperature range, there can be a debate as evidenced by the three programs being funded by DOE. MTI believes, however, that the binary steam/fluorocarbon is best in this range primarily from the point of customer acceptance. Depending upon the expected shape of the potential market distribution, the type of system to be developed would be different. Ideally, a product family with component and system commonality across the entire range would provide the greatest market potential, the lowest cost system, and the least market risk.

DISTRIBUTION OF WASTE HEAT FLOWS FOR RANKINE SYSTEMS

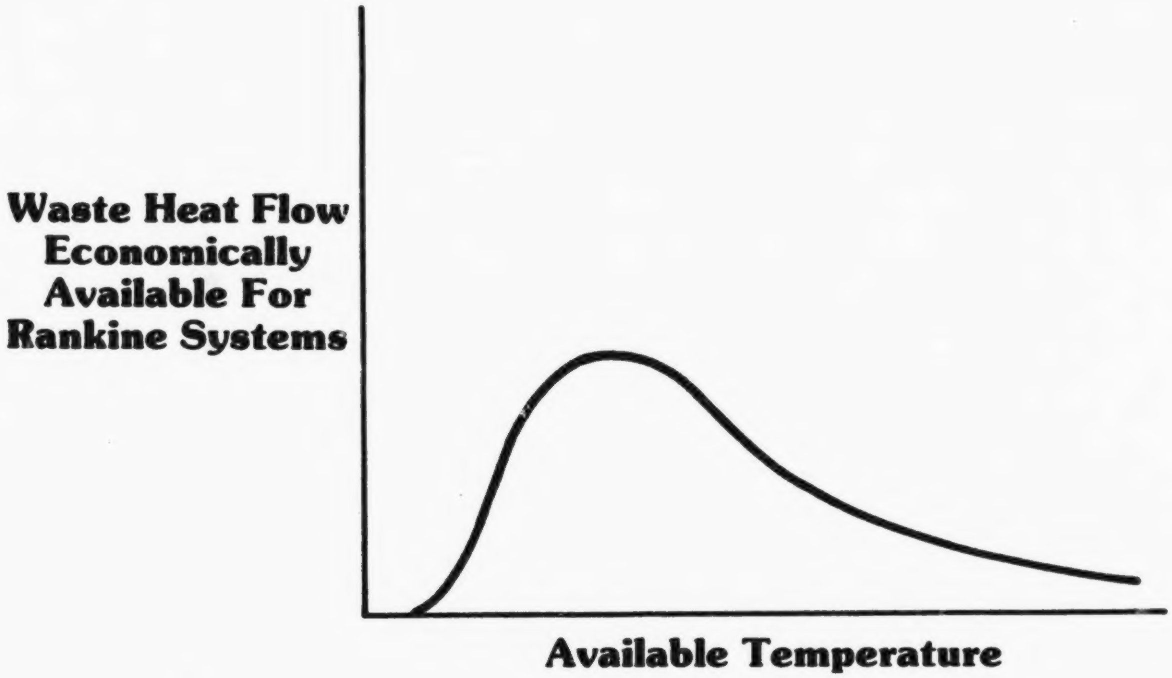

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Figure 11

RANKINE SYSTEM PRODUCT SELECTION

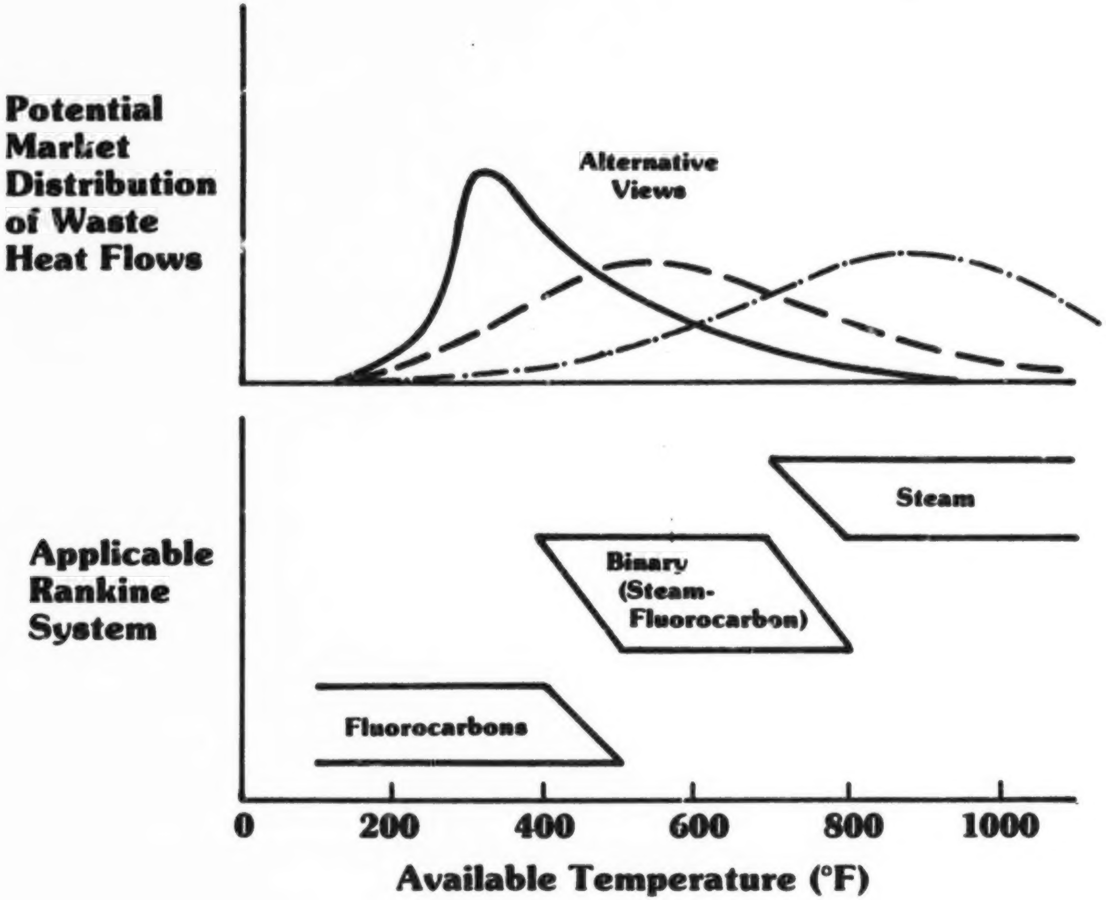

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Figure 12

4.0 USER ECONOMICS

The basic criteria for investment decisions is the rate of return that investment provides the investing company. If an analysis is made of the actual return on equity experienced by industrial and utility companies, one could determine an investment threshold. This type of exercise was conducted in an FEA sponsored study and presented here in Figure 13. There is an obvious difference between utilities and industrials. Utilities, whether publicly or privately owned, are monopolies with a guaranteed minimum return and a prohibited maximum return. Industry has more opportunity to make money as well as to lose money. With an inherent higher risk, the industrials have higher returns. The rates of return indicated in Figure 13 do not, in fact, represent investment thresholds. They represent the average return on investment. The profit motivation (as well as bonuses and stock options) forces companies to seek investments that will improve their average return. The return premium demanded depends upon the perceived risk of the investment and the type of investment. The perceived risk is a subjective or objective evaluation of the probability that the investment will achieve its performance goals. Technical and safety risks influence this valuation. The types of investment include defensive investments necessary for the company to maintain its business or market share, product expansion investments, and discretionary cost reduction investments. In-plant power generation falls into this last category. As pointed out in recent articles, industrial management has often imposed an investment return threshold that is twice that indicated in Figure 13 for discretionary energy cost reduction investment. The value of DCF shown in Figure 13 is the threshold DCF based upon the internal cost of capital for a distribution of industrial companies and utilities. The data suggests that a DCF of 20 percent will be accepted by 40 percent of all companies.

Investment in waste heat power recovery will soon become a defensive investment. Energy costs are becoming increasingly important. In some industries, fuel costs approach 20 to 30 percent of operating costs. At this point, energy cost savings will significantly affect product cost. Thus, the companies that save the most energy can reduce their selling price and obtain both a large market share and more profit. However, even in this case, the investment must show at least the returns indicated in Figure 13.

In order to evaluate an investment decision, a basic financial investment analysis must be made. This analysis is characterized by various names, such as

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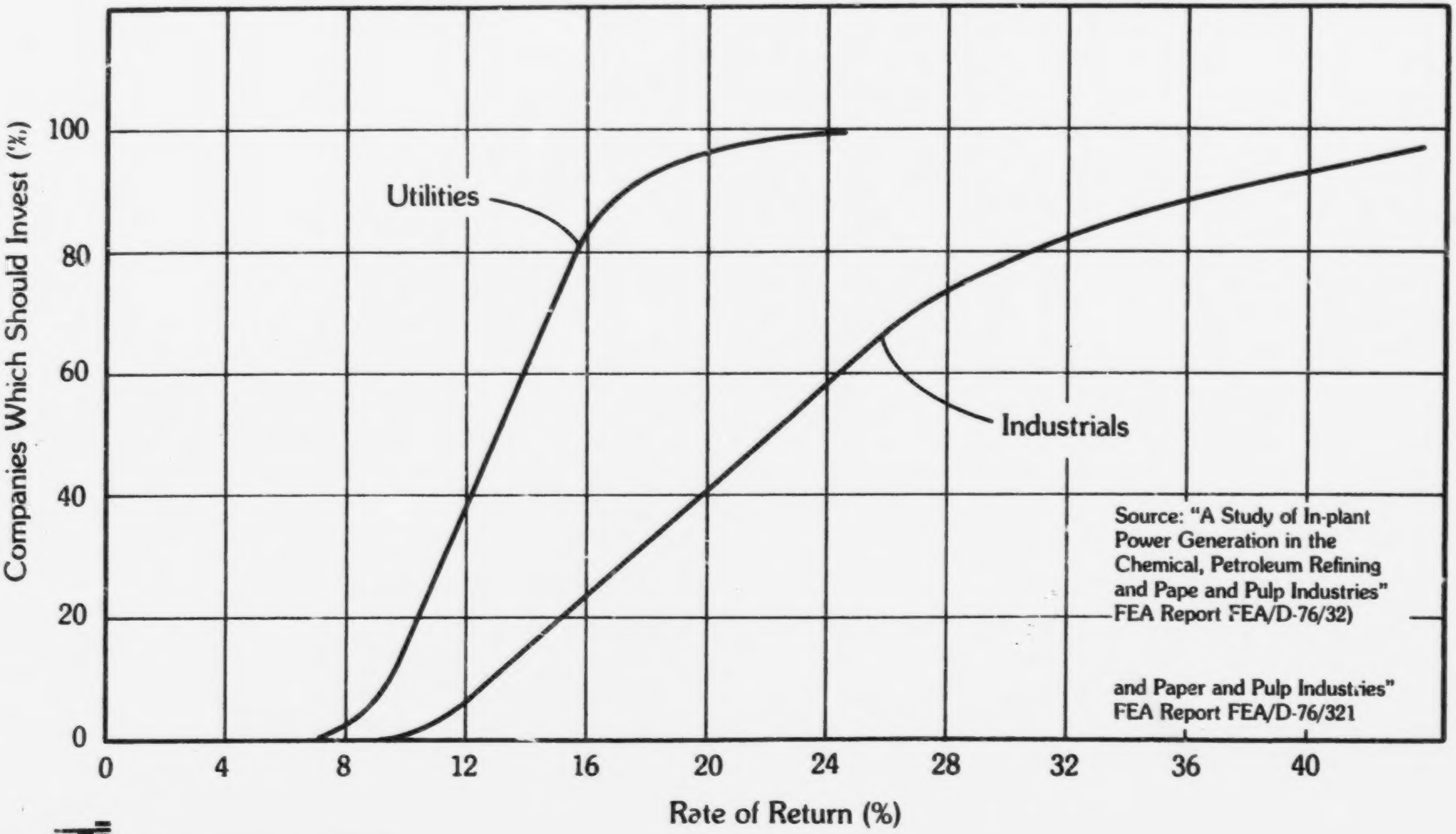
CUMULATIVE DISTRIBUTION OF RETURN ON EQUITY



Source: "A Study of In-plant Power Generation in the Chemical, Petroleum Refining and Pape and Pulp Industries" FEA Report FEA/D-76/32)

and Paper and Pulp Industries" FEA Report FEA/D-76/321

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Figure 13

Discounted Cash Flow (DCF) or Internal Rate of Return. Basically, the analysis determines a single figure of merit for the investment which is an effective interest rate, or DCF. This DCF interest rate indicates to the investor the return that investment would provide if it were a simple bond. This DCF interest rate can be compared directly to the values in Figure 13, or to alternative investments. The DCF interest rate is a function of tax rates, energy costs, inflation, depreciation rates, and investment tax credit. The next section will develop the DCF analysis, and is followed by an evaluation of the effects of the various input factors on the DCF rate, particularly the effects of governmental tax policies.

An investment analysis designed for examining investments in waste heat recovery systems was used to analyze the sensitivity of the DCF to total installed cost, the avoided electric unit cost, the energy inflation rate, the annual hours of operation, the marginal tax rate, the investment tax credit rate, and finally, the depreciation life. In this parametric sensitivity analysis, the unit price of installed machinery specified for the program is taken to include all costs including site preparation, construction, tie-in, startup and end user engineering costs. Operating costs were assumed to be negligible. The economic project life was assumed to be 10 years. Local taxes and insurance were taken as 3 percent of the initial investment.

The effect of unit price of avoided electricity is shown in Figure 14 with no inflation effects. This shows why there has not been interest in power generation from waste heat. Current average industrial electric cost in the U.S. is about 2.5¢/kw-hr. Without any inflation, the cost will remain at that level. The DCF at an installed cost of $800/kw is only 12 percent with 2.5¢/kw-hr. Referring back to Figure 13, it is clear that very few companies would invest at this level of DCF. A more reasonable and expected investment threshold would be a 20 percent DCF. The expected installed cost of a bottoming system is indicated in Figure 14 as in the range of $800 to $1,000/kw for a 1000 kw system. This implies that if a company now has electric costs of the order of 4¢/kw-hr or greater, waste heat power recovery would be an attractive investment.

EFFECT OF ENERGY COST ON THE DISCOUNTED CASH FLOW

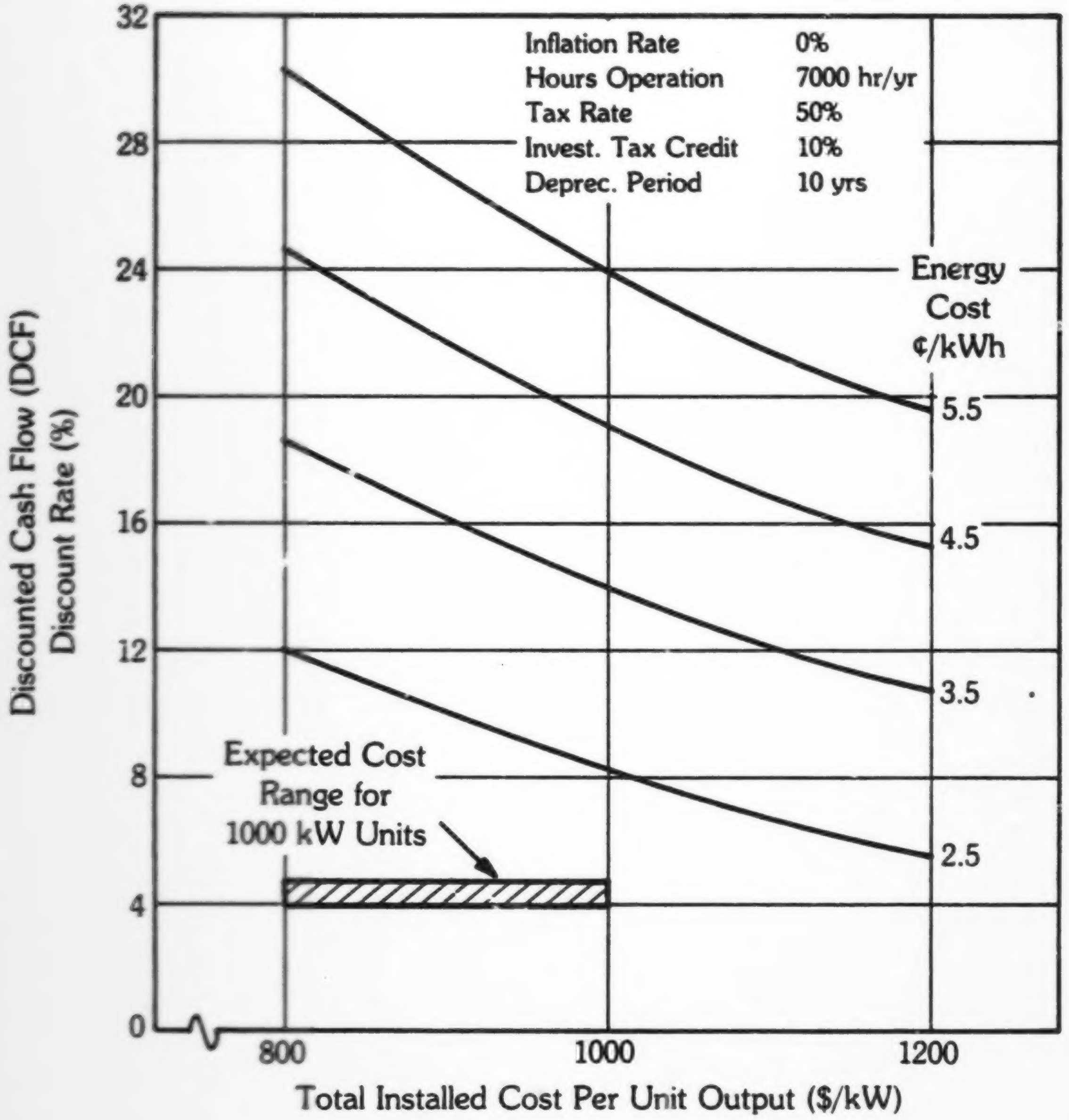

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Figure 14

In Figures 15 through 19, an area is indicated as being the region of expected near term economic acceptability for waste heat recovery systems. This region is the same for all the figures and helps to indicate the effect of the sensitivity parameter being illustrated. This area indicates the expected economic parameters that will characterize the initial commercial purchases of waste heat power recovery systems in 1979 to 1982 time frame.

Figure 15 presents the effect of electric cost inflation rate on the DCF. A variation of ± 2 percent from an 8 percent inflation rate does not have much effect (about ±1 percent in DCF). Therefore, an exact estimate of the inflation rate is not too important as long as it is a reasonable estimate. For the U.S., an electric inflation rate of 8 percent seems to be an accepted estimate.

Another parameter that has significant effect on economic acceptability is the annual hours of operation. Since waste heat recovery systems are tied to a primary operating cycle typical of a given industry, Figure 16
only those industries having high annual utilization of their plant will be attracted to use waste heat power recovery. For example, a single shift operation will have at best 2080 hours of operation in a year (40 hours/week). In order to get 7000 hours of operation, a plant has to be on a three shift operation, six days per week. This would tend to suggest a process type operation. For example, a base loaded Diesel generator has only 7500 hours per year of operation.

The parameters covered thus far are parameters that are not easily controlled or even significantly affected by the government. The next three parameters are readily controlled by the government and can be used to promote the economics of energy saving investments. Figure 17 shows the effect of the marginal income tax rate. Currently, the marginal federal income tax rate for U.S. industry is 48 percent. State taxes will be added but will be somewhat reduced by deductions in the federal tax base. Income tax rates are not an appropriate means of influencing specific investments. However, there have been some suggestions for changes in the tax rate for overall economic stimulation. Figure 17 shows that a change of 4 points in the tax rate produces about a 2 point change in the DCF. This is not a very effective stimulant for investing specifically in energy saving equipment.

EFFECT OF ENERGY INFLATION RATE ON THE DISCOUNTED CASH FLOW

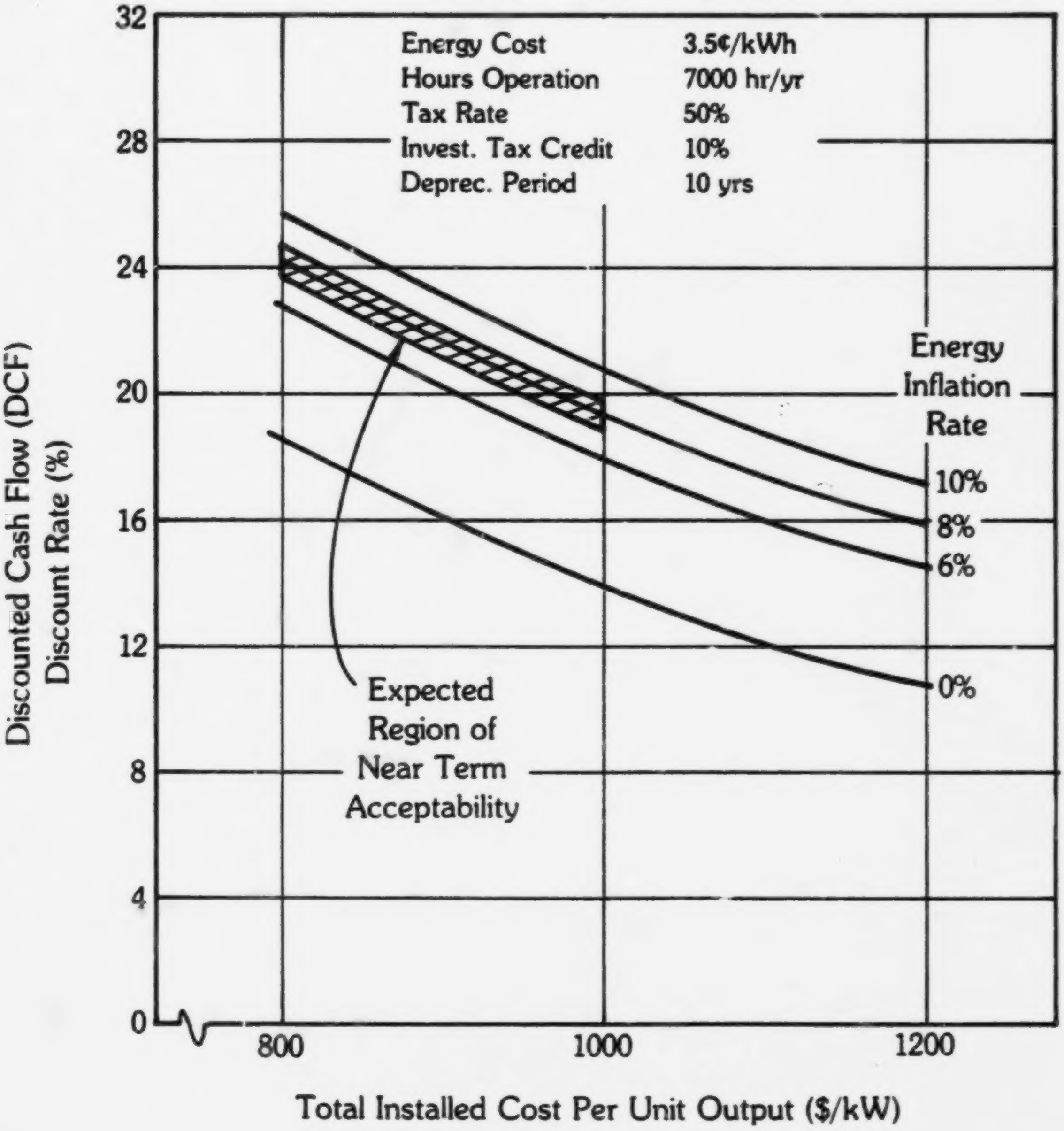

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Figure 15

EFFECT OF ANNUAL OPERATING HOURS ON THE DISCOUNTED CASH FLOW

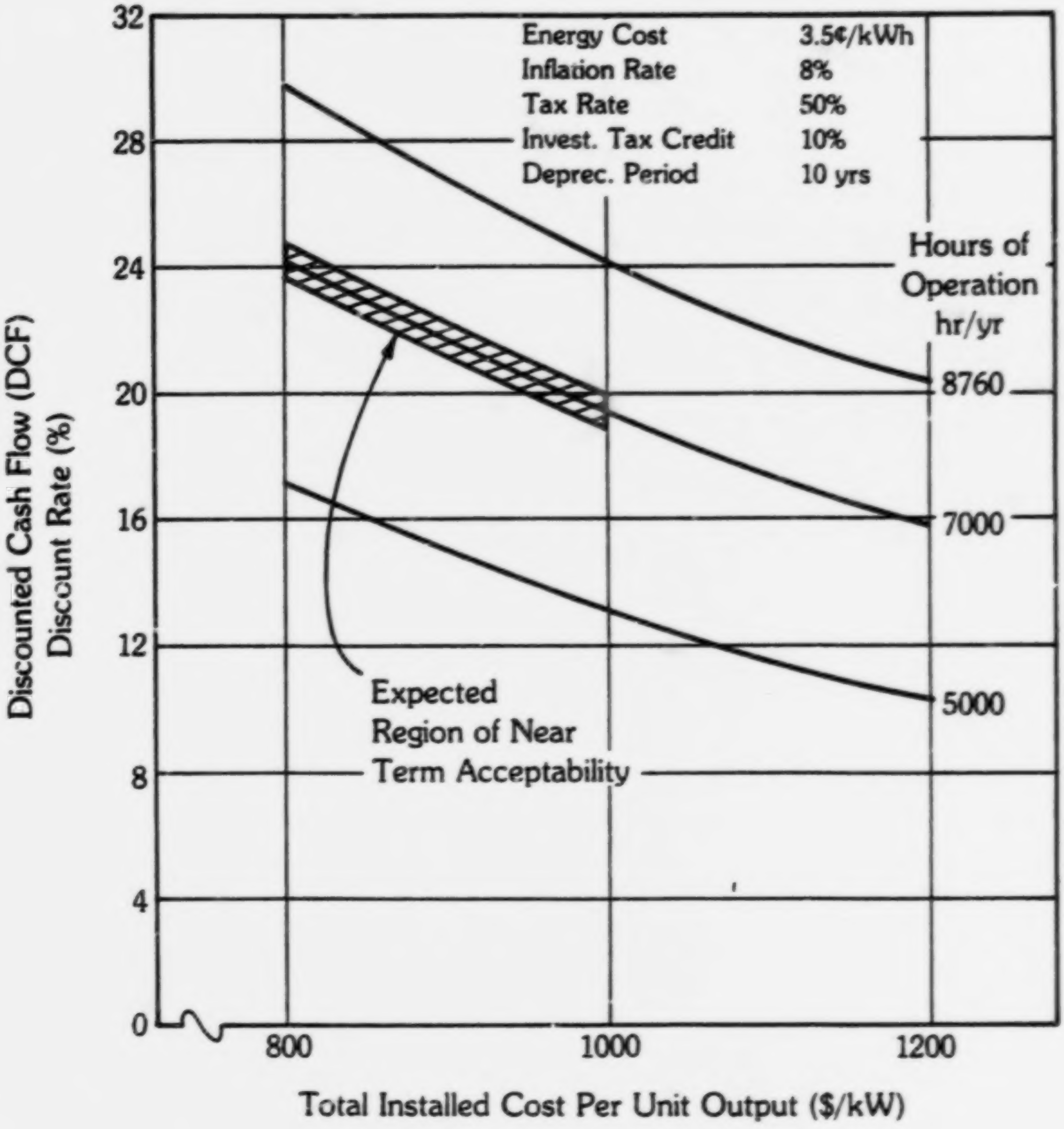

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Figure 16

EFFECT OF MARGINAL TAX RATE ON THE DISCOUNTED CASH FLOW

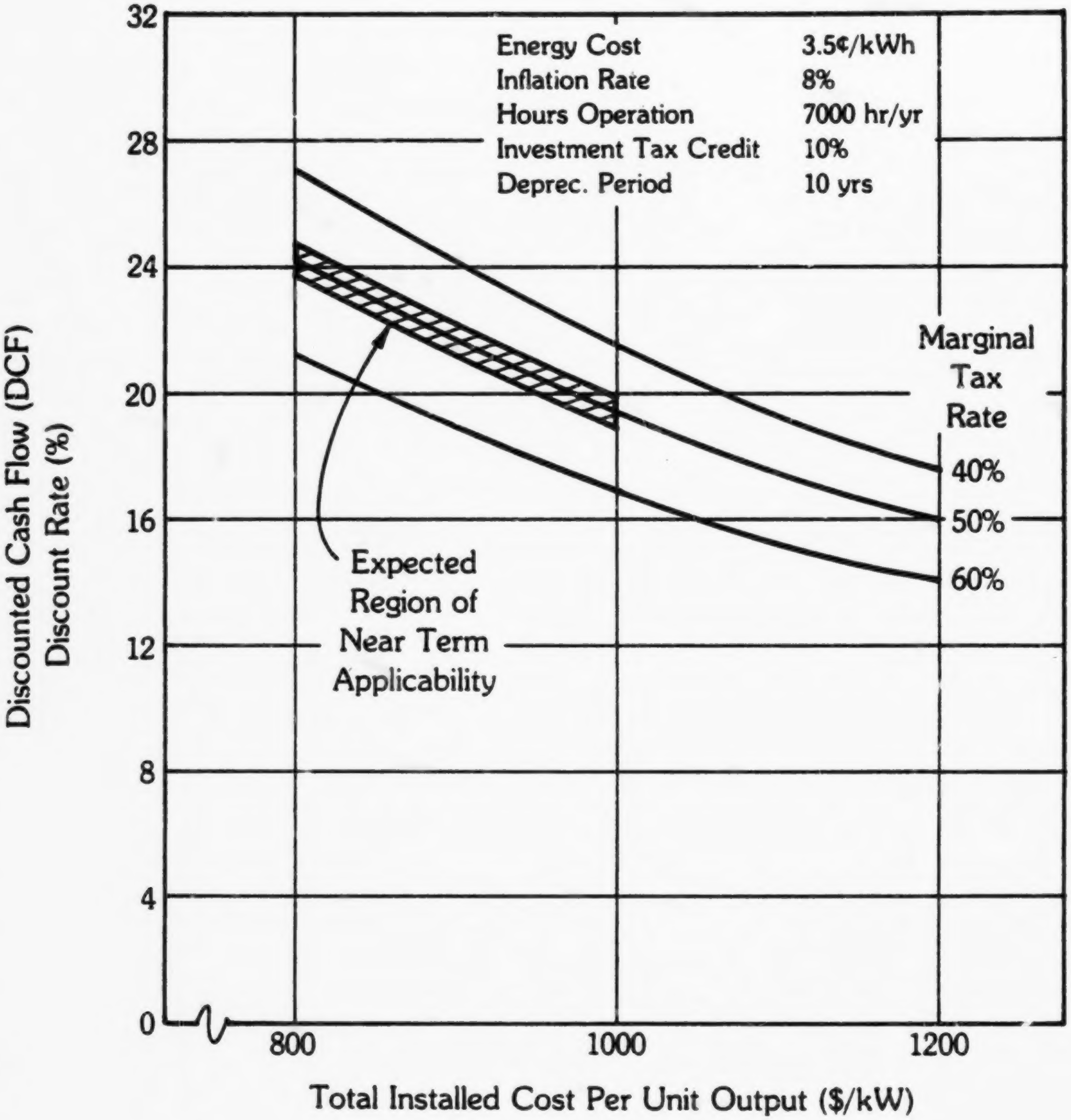

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Figure 17

EFFECT OF INVESTMENT TAX CREDIT RATE ON THE DISCOUNTED CASH FLOW

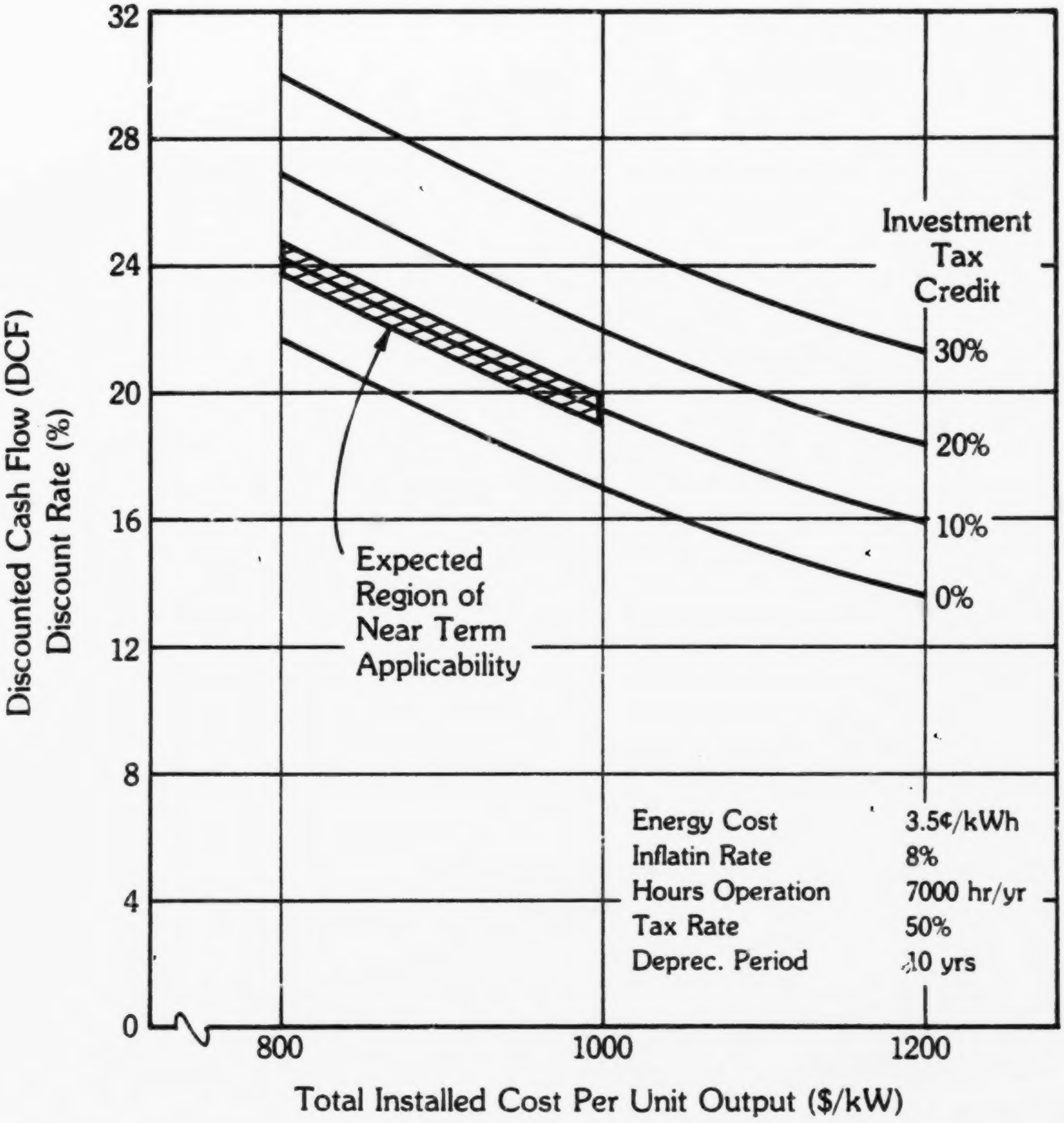

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Figure 18

EFFECT OF DEPRECIATION PERIOD ON THE DISCOUNTED CASH FLOW

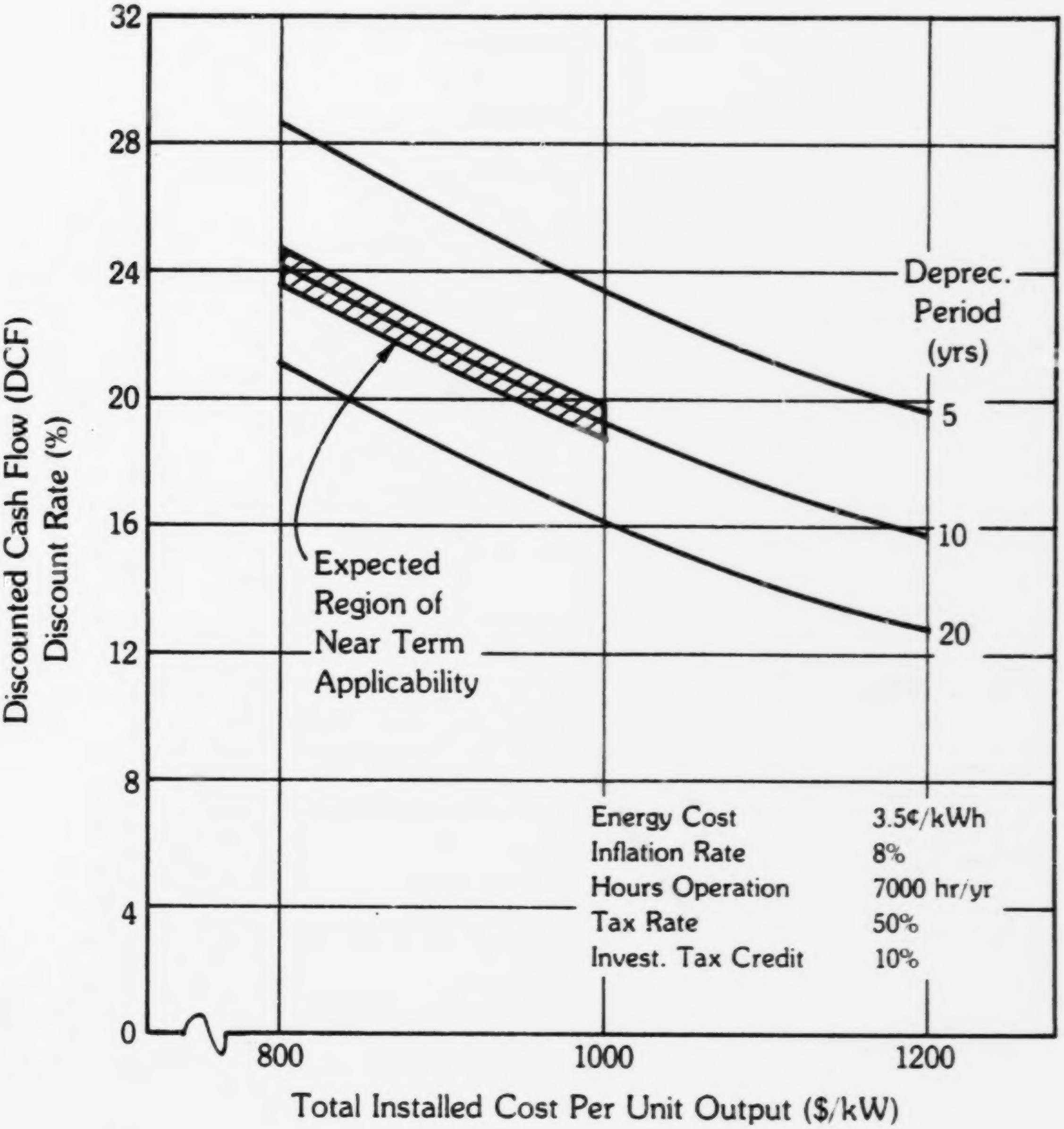

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Figure 19

Legislation has already been introduced to increase the investment tax credit for energy conservation equipment. Figure 18 shows that an increase of 10 percent in the investment tax credit raises the DCF by 3 percent. This could move a marginal investment into the acceptable region. An increase of 20 percent in the investment tax credit will raise the DCF by about 6 percent. This could move a marginal investment into the attractive region.

Figure 19 shows the effect of permitting a rapid write-off of energy conservation equipment. Reducing the depreciation period from 10 years to 5 years increased investment tax credit will significantly improve the economics of waste heat utilization technologies in the industrial sector.

The general economic analysis discussed thus far can be applied to specific Rankine cycle configurations and applications. The difference between the general economic analysis and an application of economic analysis is that the cost of a Rankine system is estimated for the application. This is accomplished by a Rankine System Design Program developed by MTI which sizes a Rankine system for a specific application and provides a cost estimate based upon component costing algorithms. To determine the installed cost, an installation charge equal to 25 percent of the machinery cost is added to the machinery cost. Results of a sample application economic analysis are presented below.

Figure 20 presents the economics of power recovery for the application of a condensing waste stream. This figure illustrates that heat recovery economics depends not only on the temperature of the waste stream, but also on the size of the waste stream. If a waste condensing stream of 30 million Btu/hr is available, a source temperature of 205°F would show a DCF of about 25 percent. This is an attractive enough DCF to warrant a more detailed look. If the heat flow were 10 million Btu/hr, a source temperature of about 300°F would be necessary to achieve a DCF of 25 percent.

This type of plot permits very rapid screening of possible heat recovery applications. Local economic conditions can be quickly taken into account with sufficient accuracy for screening purposes by using the following relation:

$$\text{DCF}_{\text{Actual}} = \text{DCF}_{\text{Figure 4-8}} \left[\frac{\text{Actual Electric Rate}}{3.5}\right] \left[\frac{\text{Actual Hours}}{7000}\right]$$

VARIATION OF DCF ORGANIC CYCLES

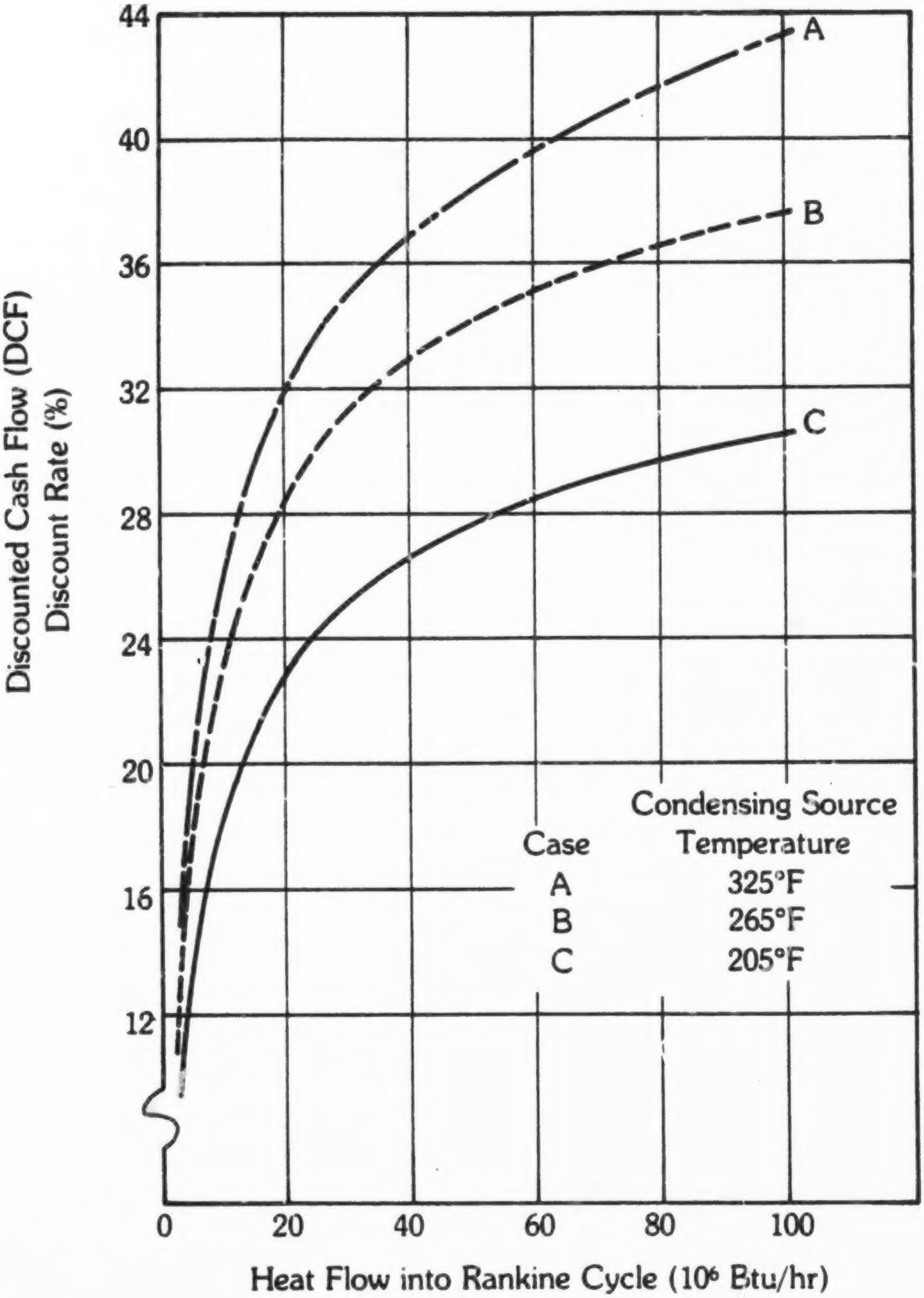

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Figure 20

Referring to the example of a 30 million Btu/hr flow at 205°F, the DCF from Figure 20 is 25 percent. If the actual electric rate is 2.5¢/kw-hr and the actual operating time is 8000 hours the screening DCF for that particular application would be about 20 percent, which is marginal considering the simplifying assumptions made for the screening purpose.

Recent discussions with a number of industrial clients indicates that a project DCF of 20 percent is needed to justify an energy conservation investment. If we use Figure 20 for screening, an extra 5 percent in DCF should be required to account for the simplifying assumptions made. If we also restrict our discussion to industries that have operations of over 8000 hours per year, we can define a screening DCF as:

$$\mathrm{DCF}_{\text{Screen}} = \mathrm{DCF}_{\text{Required}} \left[\frac{3.5}{\text{Actual Rates}}\right] \left[\frac{7000}{\text{Actual Hours}}\right]$$

or

$$\mathrm{DCF}_{\text{Screen}} = 25 \left[\frac{3.5}{2.5}\right] \left[\frac{7000}{8000}\right] = 30.6\%$$

Since this is a screening analysis, we will use 30 percent as the threshold DCF. Figure 21 presents the results showing the current source temperatures and heat flows in process industries that might be judged economically acceptable for the use of Rankine bottoming cycles. In general, applications with heat flows into the Rankine cycle of less than 15 million Btu/hr seem to be uneconomic under the criteria used in this screening.

From discussions with various companies, there seems to be a large number of applications that fall at or above the economic threshold lines in Figure 21. Thus, there is now an economic need for bottoming cycles that will increase as the energy cost increases relative to equipment cost.

What is needed to make this need into a viable business for manufactures of packaged bottoming cycle systems is actual demonstration of such systems in industrial applications.

Economic Applications for Bottoming Cycles

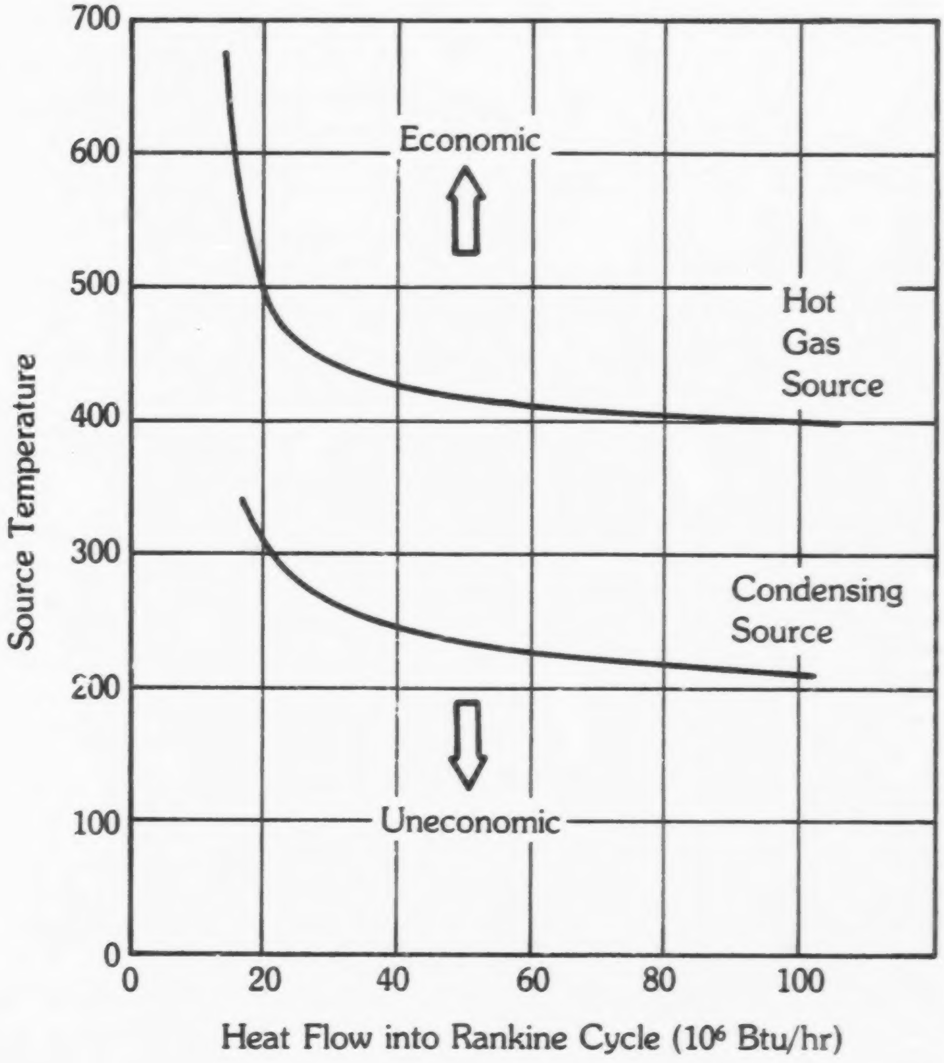

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Figure 21

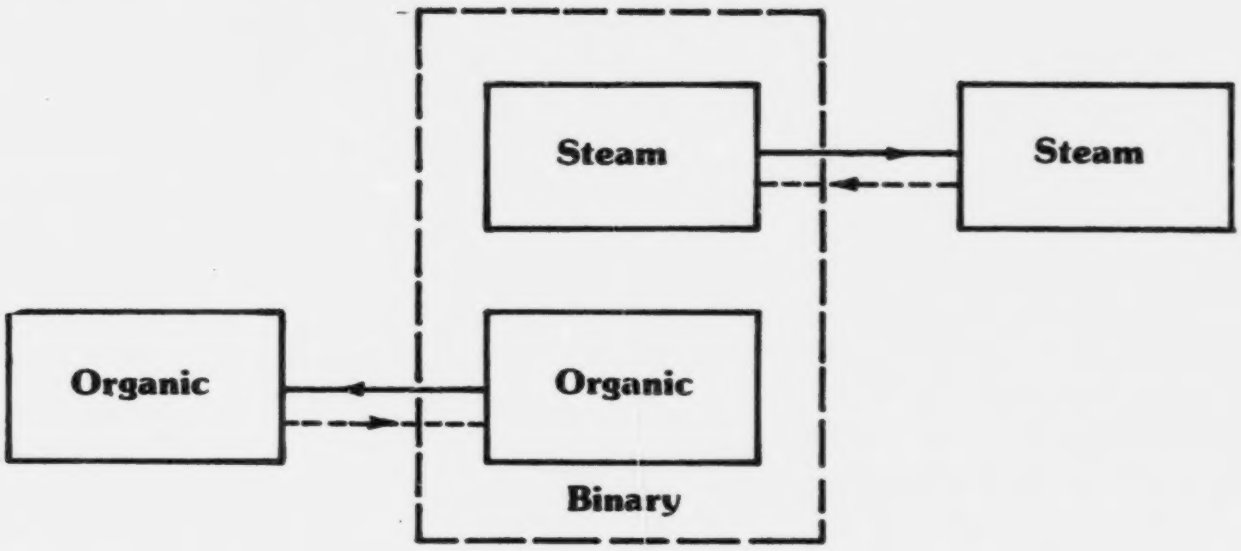
5.0 PRODUCT LINE REQUIREMENT

At this point, we have established that there is an economic basis for waste heat power recovery and that the wide spectrum of waste heat availability requires a diversified product line. The latter point is illustrated in Figure 12. This assessment and view of the potential market need has resulted in the product line concepts implicit in the DOE/MTI Cooperative Venture Program. This concept is centered on the development of a product line based upon water and fluorocarbons to serve the entire market need with as much component and system commonality as possible. This product concept was derived from the binary system developed by MTI. As shown in Figure 22, the binary system is composed of a steam cycle and an organic cycle. These individual cycles can be developed within the binary framework and then disassociated to independent cycles as shown in Figure 22. This yields three related systems to serve the entire market requirement for waste heat power recovery systems. Alternatively, advanced concepts in the steam or organic systems could be integrated together to produce an advanced and improved binary system.

The product family concept is focused to yield as low a cost system as possible by having efficiencies of scale. However, this product family concept can be totally defeated by a poor system packaging concept. It is possible to build a waste heat recovery system by ordering currently available stock components and assembling them in the field. This is shown in Figure 23 as the conventional design approach. This approach has a number of serious disadvantages. The first is that currently available components in the turbomachinery train are neither very high in efficiency nor necessarily compatible in terms of system dynamics. This would result in a number of compromises that would reduce overall system output. The second is that this approach is very expensive, engineering costs, the costs of small quantity purchases of items and, most importantly, the erection costs. The field erection cost of a system, like that shown in Figure 23, can range from 100 percent to 300 percent of the base hardware costs. These two factors will increase the unit cost to an uneconomic level. Furthermore, this approach would be very difficult to justify in terms of reliability and performance.

The package approach taken by MTI is that of a highly integrated modular design. As shown in Figure 24, only three modules are delivered to the field and this

MTI RANKINE SYSTEM PRODUCT LINE CONCEPT


Steam
Steam
Organic
Organic
Binary

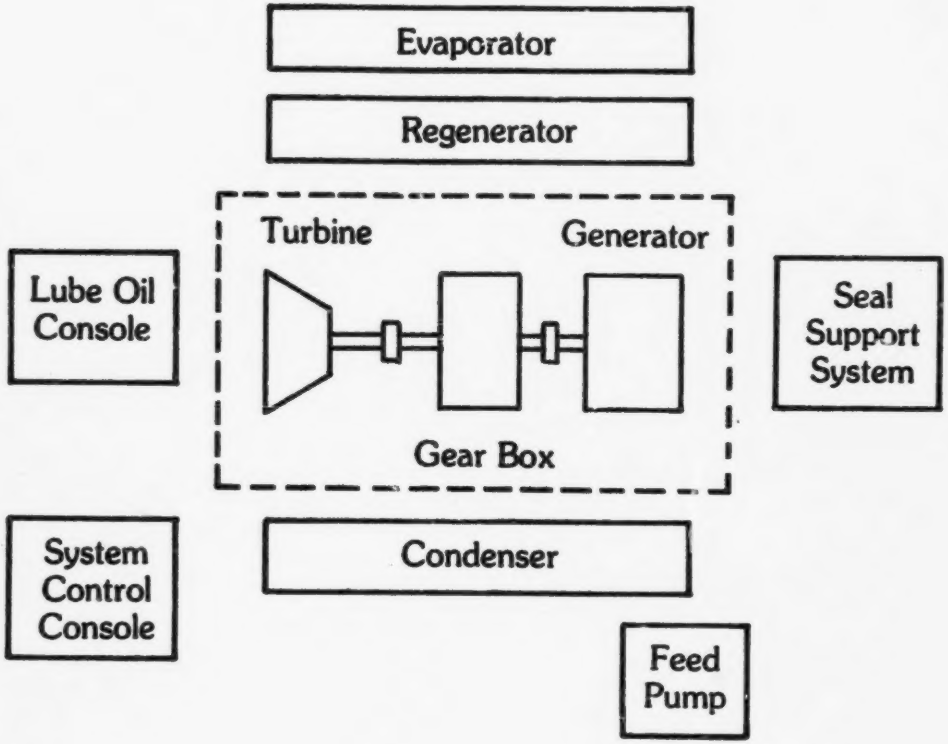
Source:	250 - 500°F	400 - 800°F	700°F Up
Size:	500 - 2000 kW	500 - 2000 kW	1000 - 4000 kW
Application:	Process Streams	Diesels, Stacks	Gas Turbines, Stacks

MECHANICAL TECHNOLOGY INCORPORATED

Figure 22

CONVENTIONAL SYSTEM DESIGN APPROACH

- ☐ Separate Machinery Components
- ☐ Costly Field Interconnections
- ☐ High and Low Speed Couplings in the Field
- ☐ Maximum Space Required
- ☐ Numerous Seals with Attendant Leakage
- ☐ Extended Field Erection
- ☐ Start-up and Check-out Done in the Field



Evaporator

Regenerator

Turbine

Generator

Lube Oil Console

Seal Support System

Gear Box

System Control Console

Condenser

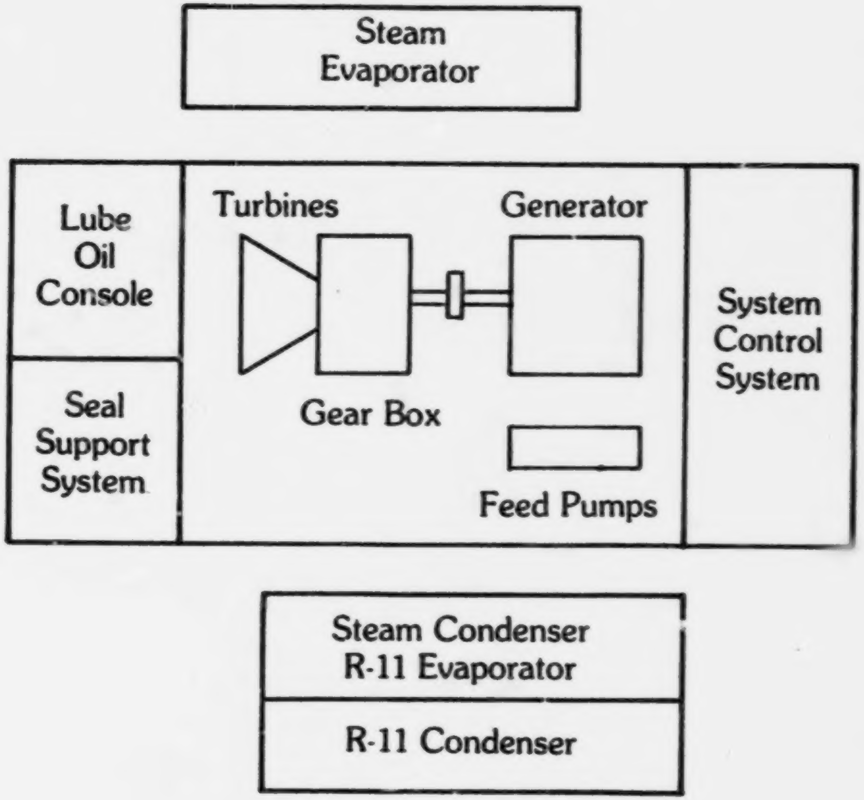
Feed Pump

MECHANICAL TECHNOLOGY INCORPORATED

Figure 23

MTI/TURBONETICS SYSTEM DESIGN APPROACH

- ☐ Unitized Machinery and Control Construction
- ☐ No Field Interconnections for Machinery System
- ☐ No High Speed Coupling
- ☐ Reduced Space Requirements
- ☐ Reduced Seal Requirements and Leakage
- ☐ Reduced Erection
- ☐ Machinery Started-up and Checked-out at Factory



Steam Evaporator

Lube Oil Console

Seal Support System

Turbines

Generator

Gear Box

Feed Pumps

System Control System

Steam Condenser R-11 Evaporator

R-11 Condenser

MECHANICAL TECHNOLOGY INCORPORATED

Figure 24

minimizes field erection costs considerably. Also, the turbomachinery is designed for high efficiency and component compatibility. The system performance can be checked out prior to shipment. The integrated design approach can be subjected to better reliability testing with a single point responsibility for system reliability. This approach is currently being used with a high degree of success in the compressors currently manufactured by MTI.

6.0 COOPERATIVE AGREEMENT

The DOE/MTI Cooperative Venture Program is a 60 month, $7,788,000 effort to demonstrate the commercial viability of the product family concept for the Waste Heat Power Recovery Systems. The program is cooperatively funded with MTI absorbing $2,010,000 of the program costs. The objective of the program is to design, build, test and demonstrate all the components of the product family. Specifically, an organic, steam, and advanced binary system will be developed and demonstrated in industrial settings. The program also includes a fourth system which will be an engineering unit test bed for system and component development and evaluation.

The systems to be developed under the Cooperative Venture Program are expected to push the state-of-the-art and be advanced in performance, cost, and system integration over prior efforts including the MTI Binary System at Rockville Centre. We are not prepared, at this time, to elaborate on the specific design features of each system. However, the organic system will be based upon a novel turbomachinery concept which will significantly improve system reliability and yet reduce cost. The steam system will be based upon a multi-level steam pressure cycle with high efficiency turbines in an integrated package. The potential of extraction steam is also being considered. The binary system will derive from both the organic and steam system advances plus some system design features applicable only to the binary concept.

The key milestones for the Cooperative Venture Program are shown in Figure 25. From an industrial viewpoint, the end of 1978 is a key milestone. It is anticipated that the demonstration sites for the three units will have been selected by the end of 1978. The actual demonstration installations will occur in 1980 and 1981 as shown in Figure 25. The arrangements for participation in the demonstrations are somewhat negotiable subject to DOE approval. It is expected that each demonstration site will pay an economic value for the equipment installed.

Cooperative Venture Program

- Schedule: 60 Months (Start 4/78)

- Funding:

DOE	$5,778,197
MTI	2,010,000
Total	$7,788,197

- Demonstration Unit:

Advanced Organic .	1000KW
Advanced Steam ..	1500KW
Advanced Binary ..	1000KW

- Engineering Unit:

Advanced Binary ..	1000KW

KEY MILESTONES OF DOE/MTI COOPERATIVE VENTURE PROGRAM

Task	Calendar Year				
	1978	1979	1980	1981	1982
Engineering Unit		◆ Begin Engineering Tests			
Demonstration Site Selection		◆ Sites Selected			
Organic Demonstration			Initiate Testing ◆ ◆ Installation at Demo Site		
Steam Demonstration			Initiate Testing ◆	◆ Installation at Demo Site	
Binary Demonstration				Initiate Testing ◆	◆ Installation at Demo Site

MTI MECHANICAL TECHNOLOGY INCORPORATED

Figure 25

7.0 CONCLUSIONS

DOE and MTI believe that Rankine Cycle Waste Heat Power Recovery Systems are economic and necessary for the nation's and industry's goals of energy conservation. MTI believes that its approach of using conventional and acceptable working fluids in a product family design to serve the full range of industrial waste heat streams is the best market approach. MTI is currently completing its design specifications for that product family and will shortly be approaching industry to select demonstration sites. We expect and have already seen a great deal of interest in participating in the demonstration program. Whether or not an individual company or plant site participates in the demonstration, it will derive benefits from the demonstration and will have the opportunity to purchase additional production systems.

Remarks by
K. J. Brondyke
Director, Alcoa Laboratories
Aluminum Company of America
before the
Engineering Foundation Conference
Henniker, New Hampshire
1978 August 15

I'd like to start today by offering a quick overview of the aluminum production process, since I'll be talking about energy in each of these operations.

Our production process has four distinct segments. First, we mine and dry the ore, which today is bauxite. We then refine the bauxite into aluminum oxide, or alumina. The next step is smelting, where the oxide is reduced to aluminum and oxygen, and finally, we fabricate the metal, most often after alloying, into a variety of products that are sold to others who convert them into end products.

There is no question that the manufacture of aluminum is energy intensive. And since we use large amounts of energy, it is one of our major costs and, therefore, it represents a significant cost reduction opportunity. I am going to describe some of our efforts in capturing and reusing waste heat. But first I want to describe what is by far our largest energy-savings opportunity -- the smelting processes.

While energy is required for each step, from mine to finished goods, 70 percent of the total energy Alcoa consumes is used in the smelting process. And although it is the electric current that pries these close companions apart, a great deal of heat is generated, both in this process and in the preparation of the carbon anodes and cathodes used in the cells.

How much of this heat is "waste" is a matter of definition. What may be surplus to one manufacturing process may be vital to another. For instance, the present state of materials technology requires some waste heat in our smelting operations. Let me explain.

Even though there is a tremendous amount of heat generated, the corrosive nature of the molten salt bath requires that we design our smelting cells to allow a major portion of this heat to escape the cells unused. Only in this way can we freeze the bath on the sidewalls, creating a solid barrier between the side wall refractory and the molten salts. If we didn't design this inefficient heat transfer capability into our cells, we simply couldn't operate. If any of you want to get rich, just tell us how to make a smelting cell that will survive without a frozen bath layer.

Improving energy efficiencies is a prime objective of both Alcoa and the aluminum industry. At Alcoa Laboratories, in addition to efforts to produce molten aluminum as economically as possible, we also must determine the most desirable mechanical and metallurgical properties of our fabricated products and find the most energy efficient methods for producing metal having these properties. This is the technological approach, and we feel it is economically justifiable.

I'd like to review some of our major projects aimed at improving Alcoa's mining, refining, smelting and fabricating processes. We regard our entire production process as a totally integrated system. Therefore, when we begin to look for energy-saving ideas, we begin with the first operation, mining . . . primarily, bauxite and fossil fuels.

In the last year, we have worked out a computer program that enables us to develop our mining on a scientific basis. With this computerized approach, the prospective fossil fuel or aluminous ore mine is partitioned into ordered blocks. By knowing the composition of the ore in each block, the program allows selection of "best order" mining. In this way we are better able to provide a homogeneous plant feed and optimize energy efficiency of the operation.

While smelting consumes 70 percent of our total energy, not all of that consumption occurs in the smelting cell. We use a mixture of carbonaceous materials for anodes and cathodes. In the preparation of anodes, a great deal of energy is used in the curing -- or, as we say, baking -- process. Last year, after a thorough study of heat loss, in our ring baking furnaces, we installed specialty insulation that reduced heat loss from air in-leakage. As a result, we are now enjoying a 13 percent energy savings in these furnaces. When we have insulated all our ring furnaces, we estimate a 350 billion BTU savings per year. Incidentally, that translates into a savings of around $700,000 a year at today's prices for gas.

Using data from our thermal study we also were able to come up with a new ring furnace design that will cut present fuel use by 50 percent. By going to bigger, deeper pits and tighter seals, we are saving 280 billion BTU's per year. That's a cost savings of more than half a million dollars -- each year.

Moving to the smelting cell itself, one of our major projects, currently in the pilot stage, uses refractory hard metals as the cathode to reduce voltage loss. Kaiser Aluminum, in a government program, also is exploring this area. If these programs are successful, they will offer the industry the pleasant option of either reducing power consumption while sustaining metal production or increasing our output without using more energy. A factor in this increased efficiency is reduced heat generation per pound of metal produced.

Over the years, we also have undertaken extensive programs to improve the characteristics of the molten salt bath that acts as the solvent for aluminum oxide in the reduction process. This involves additives, largely lithium and magnesium fluorides, to lower the melting point of the bath. This permits lower operating temperatures, which reduces heat loss and improves ampere efficiencies.

Dramatic advances in computer technology, coupled with our improved understanding of smelting cell dynamics, have paved the way for two programs that we feel have major implications for the future operation of our smelters. These computer programs marry our knowledge of operating variables with never-before-available information provided by improved sensing devices. We have modeled the cell and can analyze changes in individual cell operations and even entire smelting plants very quickly. We can do in a matter of hours or days what used to require months. When totally implemented, we expect significant benefits from these innovations.

Actual computer control of the individual cells is being achieved through newly developed heat flux sensors. Better control reduces temperature fluctuations that contribute to heat loss. One of the largest temperature disturbances in the cell is an anode effect, caused by too little alumina in the electrolyte. When this happens, the cell voltage will rise nearly sixfold, there won't be any aluminum produced, and essentially all the power will be consumed as waste heat.

On the other hand, too much alumina ends as undissolved sludge. The result is increased electrical resistance that causes the cell to overheat. Close control will make it possible to feed the proper amount of alumina and prevent either of these energy-wasting conditions from developing.

After voltage losses have been minimized and the cell control fine-tuned to the best of our ability, we will turn our attention to concepts for recovering waste heat for use either in other processes or conversion to electrical energy.

In our operations, the benefit/cost ratio of recovering waste heat from the cells is much lower than that of reducing its generation. Consequently, our effort in smelting has been directed toward generating less waste heat rather than trying to recover it.

The industry's interest in reducing the energy to smelt a pound of aluminum didn't start with the energy crunch of the 1970's. It has been an historical one. The greatest proof of this long-time interest is the development of the Alcoa Smelting Process.

We began working on this aluminum chloride method to produce aluminum in the late 1940's. Twenty-three years and 25 million dollars later, we patented the first major development in aluminum reduction in nearly a century. It's doing a good job. The aluminum industry today averages 8 kilowatt hours per pound of aluminum produced. Alcoa's company-wide average is less than 7.5 kwh. Our unique Alcoa Smelting Process cells have produced metal consistently at 4.5 kwh per pound. We have proven the process can do what we thought it could, and the operation is being expanded.

While we have no current programs to recover waste heat from our smelting operations, we have been busy in other areas of production. We have a number of projects under way to evaluate the different types of commercially available recuperators. In melting, we are taking a close look at ceramic heat wheels and radiation recuperators. In the pre-heat and annealing areas, we are evaluating both metallic and ceramic plate and plate-fin-type recuperators.

In addition, we've been looking into the utilization of hydrocarbons from scrap coatings as a supplemental fuel source in scrap melting operations. We also have developed a furnace design that uses a radiation recuperator to preheat combustion air and then, subsequently, uses the stack gases in two other proprietary methods to heat the charge. We've reached a cumulative heat efficiency of 42 percent. With further work, we think the potential is at least 50 percent.

Furnace burners continue to be a real tough nut. Most burners were developed to heat -- the faster the better -- and never mind fuel inefficiencies. When you add Alcoa's goal of dual-fuel capability -- to burn oil or gas -- to the need for greater fuel efficiency, the design problems multiply. Commercially available burners will have to be redesigned to withstand higher temperatures and permit switching from one fuel to another.

Alcoa's production equipment labs have been working with a number of commercial burner manufacturers to come up with a more efficient, dual-fuel burner. Some of our redesign ideas have been incorporated. I feel one of our most difficult tasks will be to find just the right materials and designs to meet the complex and stringent demands for metal heating applications.

Alcoa's Laboratories also are working on ways to develop more complex control systems to compensate for air temperature changes and maintain optimum fuel-to-air mass ratio for efficient combustion. Increased combustion gas temperatures may exceed existing safe material operating temperatures for many furnaces, particularly where combustion occurs in radiant tubes.

The aluminum industry relies heavily on radiant tube heating. Due to the reactive nature of aluminum, we have gone to radiant tubes to limit both high-temperature oxidation (HTO) and absorption of hydrogen into the metal. Hydrogen occurs as a reaction product of aluminum and water in the combustion atmosphere. HTO will contribute to fabricating problems and hydrogen can have an adverse effect on properties.

Most of Alcoa's furnaces for preheating and annealing use this indirect heating concept. The metal is heated by air which is heated by gas-fired radiant tubes. In other words, the combustion gas products don't come in contact with the metal. We have traditionally gone to this muffled effect because we were willing to sacrifice fuel efficiency in order to maintain the integrity of the fabricated product. I've had no problem with this trade-off, especially since most of the sophisticated products we treat in this way are for the aircraft industry . . . I have no trouble especially when I'm flying.

We are experimenting with direct firing today. We know that bringing the products of combustion into direct contact with the metal is a more energy efficient system. We have to be certain of the conditions under which we can go to direct firing while maintaining the metallurgical standards required by our customers.

In another furnace related project, we have installed air curtain dampers on the 90 holding furnaces throughout the Alcoa system. This project is providing a 17 percent fuel savings at a relatively low installation cost.

Finally, and of great importance to the aluminum industry is the extraordinary opportunity for increased energy savings and waste heat utilization through recycling. Recycling saves 95 percent of the energy required to produce molten metal from bauxite.

Alcoa -- and the aluminum industry overall -- is aggressively pursuing the reclamation and recycling of so-called old scrap -- post-consumer scrap. We have always recycled our own process or runaround scrap, and new scrap, which is clean scrap from customers. Today, it's important also to reclaim all the post-consumer scrap we can, to save energy and provide additional metal supply. Our current annual level of aluminum beverage can reclamation is approaching the output of an average size potline. This is equivalent to 1.4 million barrels of oil.

We believe it's essential to develop technology to reduce metal losses in the recycling process. Ten years ago, before recycling became the essential source of new metal it is today, scrap melting was a labor intense, environmentally unacceptable process done on a small scale, primarily in small foundries and by secondary smelters. Melt losses ranged from 6 to 12 percent.

As the major aluminum producers began to get in the act, new technology was developed to reduce manpower, increase productivity and improve the environmental aspects of the operation. Unfortunately, melt losses skyrocketed when this was done . . . into the 20 to 25 percent range.

Today, with Alcoa remelting about 100 million pounds of scrap per year, a one percent reduction in melt loss will net an additional million pounds of metal. It also nets an additional 100 billion BTU's of energy we won't have to use to produce virgin metal. Again, that's only a one percent reduction in melt loss. We feel melt losses in the two to three percent range are technically feasible.

Some promising, experimental work is underway at Alcoa Laboratories using dual-hearth furnaces and special delacquering and deoiling techniques. Successful new remelt technology also is reducing the amount of virgin metal added to the molten metal before producing new can sheet. We're working on literally scores of R&D projects to reduce energy requirements and improve efficiency.

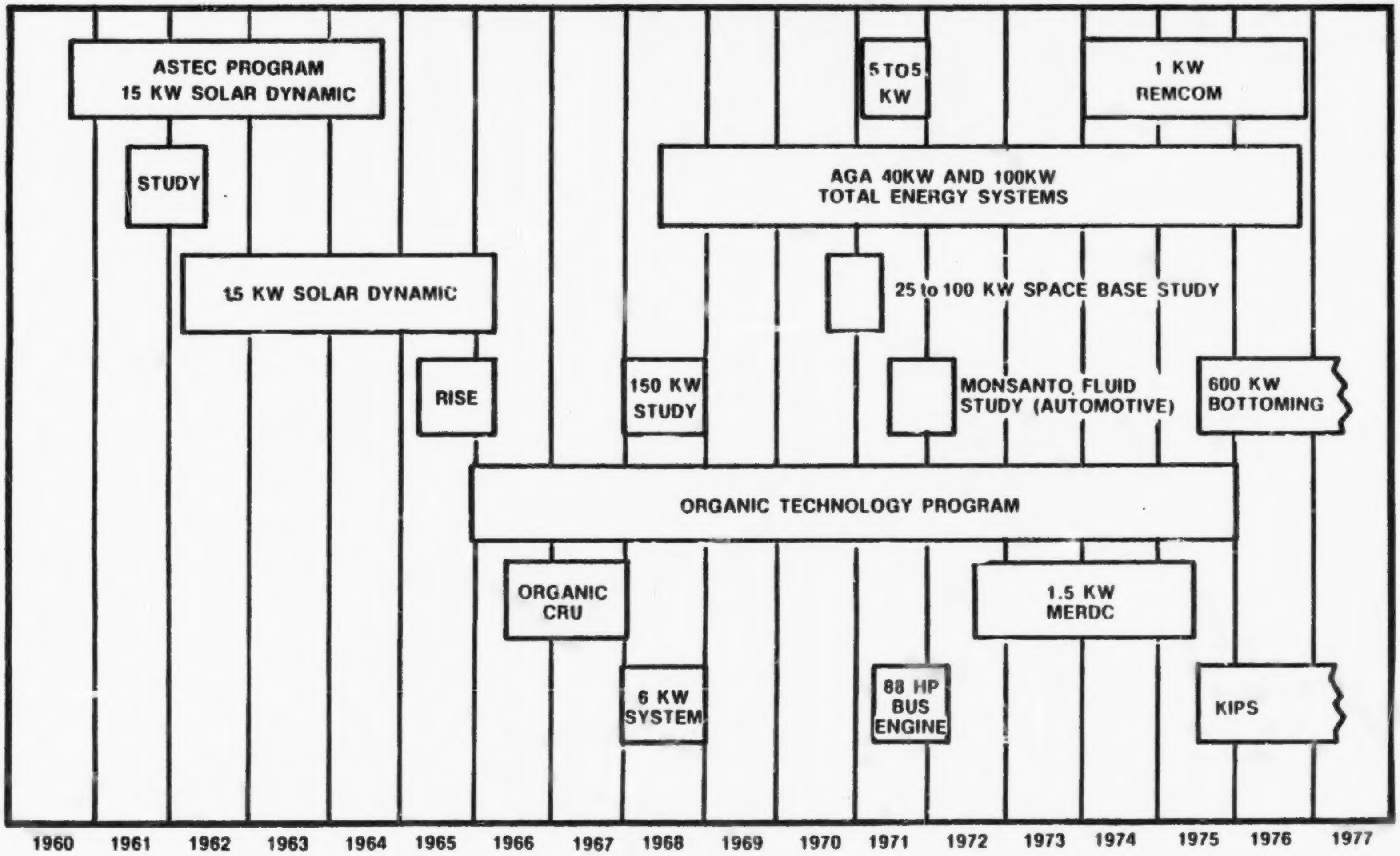
As I mentioned at the outset, our technological goals are established by the metallurgical and mechanical qualities demanded of our product by our customers. With these standards in mind, we set about determining the most efficient and economical methods of producing these products. With energy a major factor in our cost-benefit evaluations, we have been, and will continue, to do everything feasible to make the best possible use of every BTU.

600KW
ORGANIC RANKINE CYCLE
WASTE HEAT POWER CONVERSION SYSTEM

Sundstrand Energy Systems
unit of Sundstrand Corporation

106

SUNDSTRAND ORGANIC RANKINE CYCLE PROGRAMS



ASTEC PROGRAM
15 KW SOLAR DYNAMIC

5 TO 5 KW

1 KW
REMCOM

STUDY

AGA 40KW AND 100KW
TOTAL ENERGY SYSTEMS

15 KW SOLAR DYNAMIC

25 to 100 KW SPACE BASE STUDY

RISE

150 KW
STUDY

MONSANTO, FLUID
STUDY (AUTOMOTIVE)

600 KW
BOTTOMING

ORGANIC TECHNOLOGY PROGRAM

ORGANIC
CRU

1.5 KW
MERDC

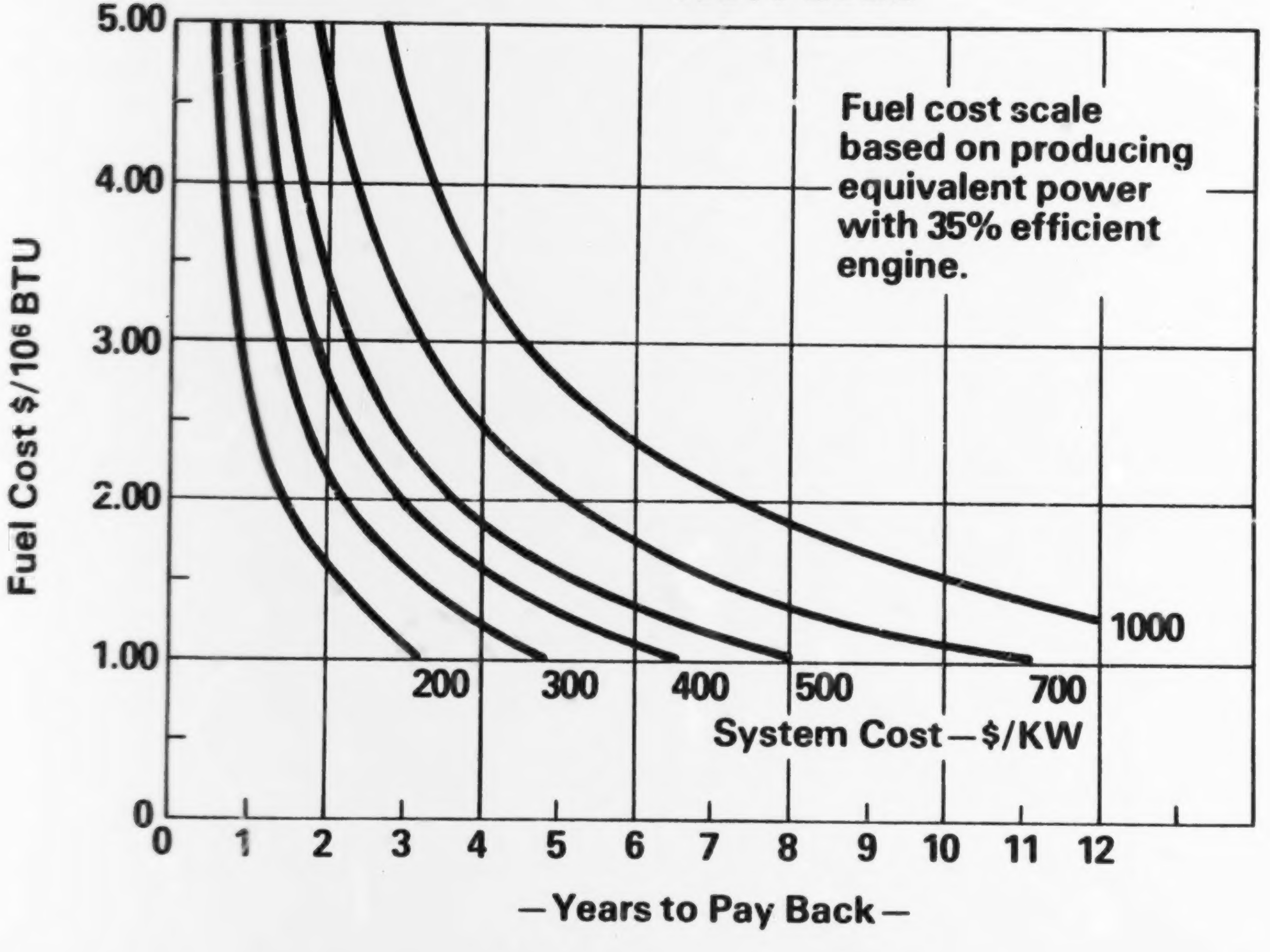
6 KW
SYSTEM

88 HP
BUS
ENGINE

KIPS

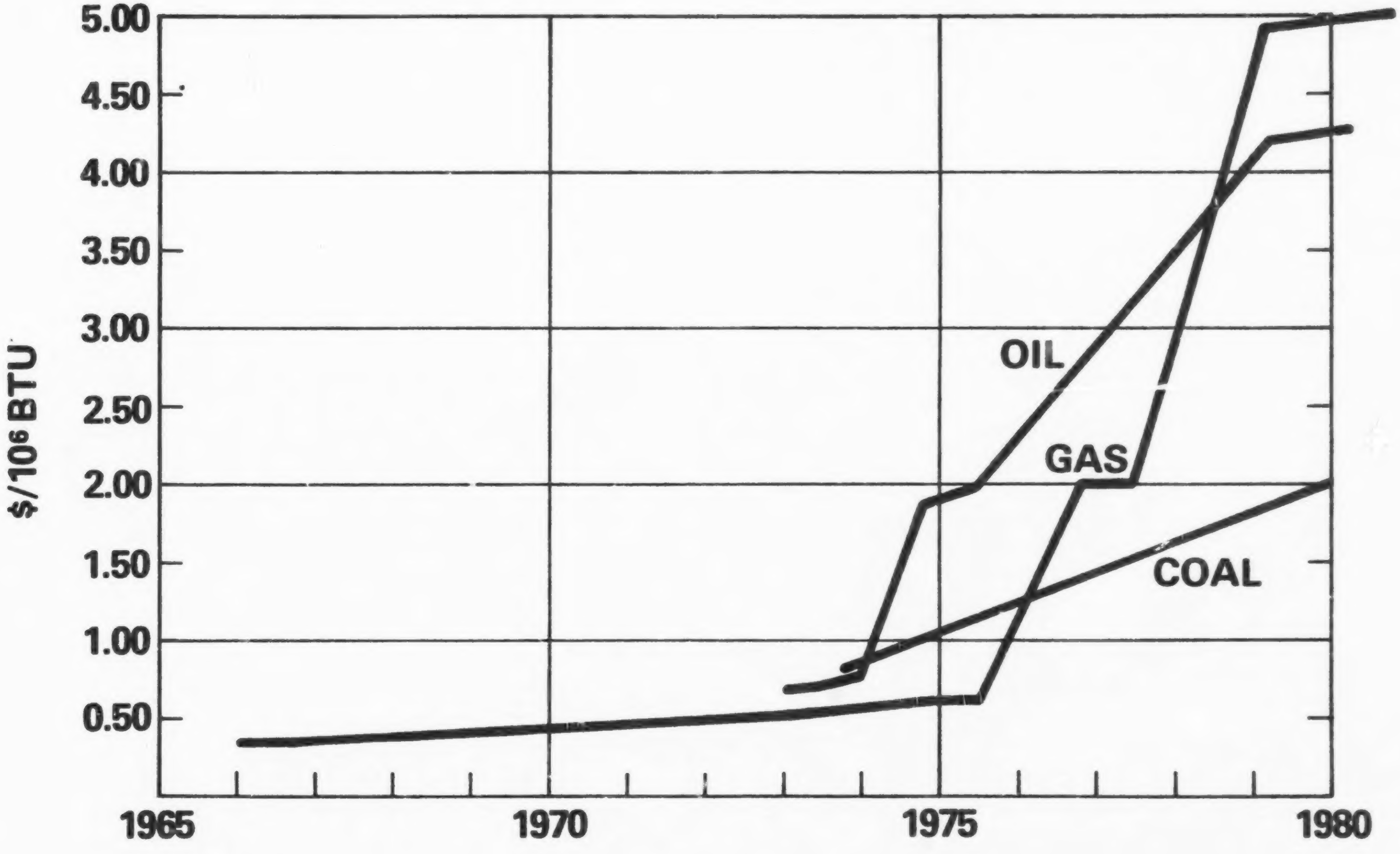
BOTTOMING SYSTEM PAY BACK

ASSUMPTIONS: 8000 Hrs/Yr Operation
100% Load



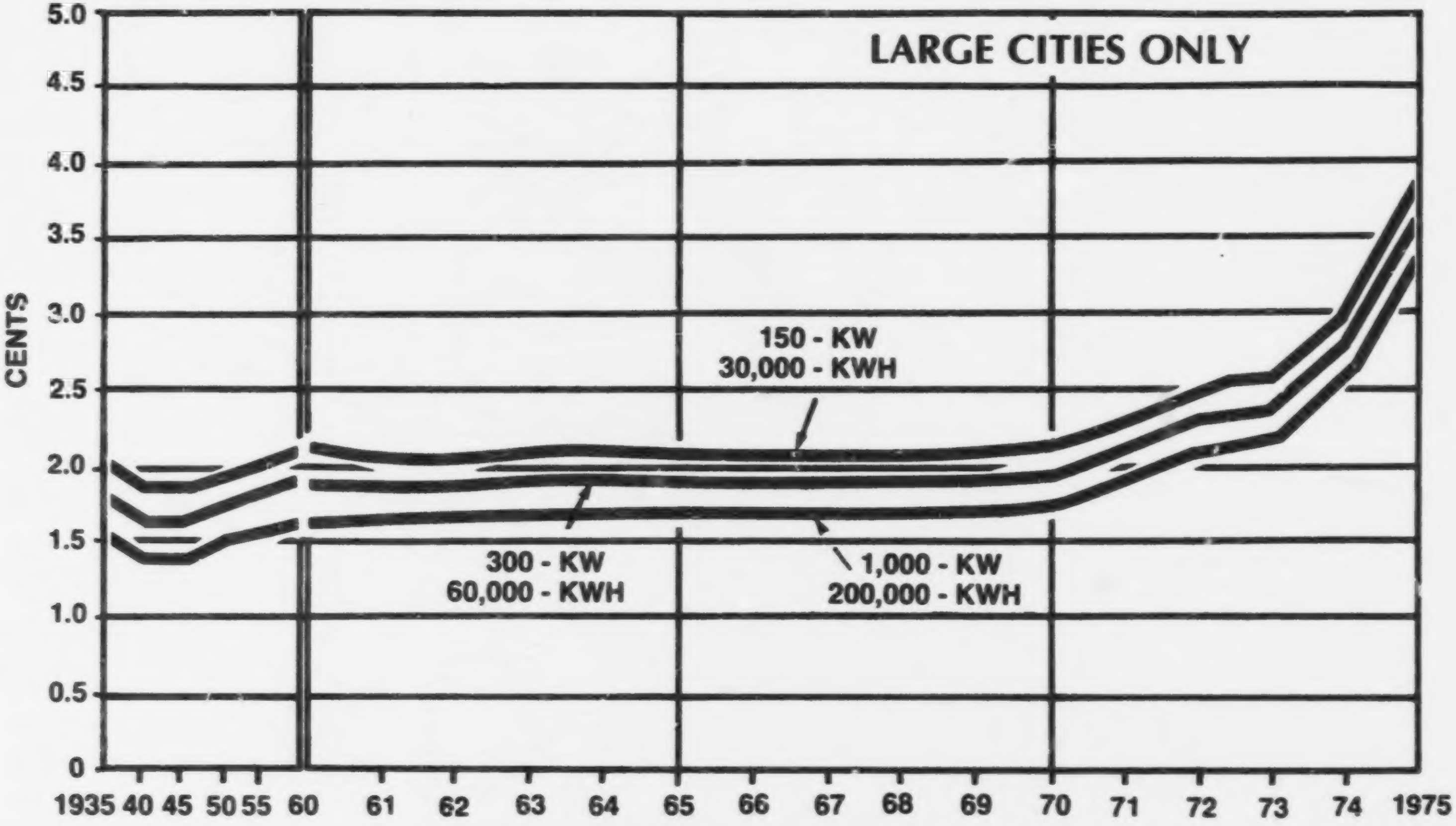
Fuel cost scale based on producing equivalent power with 35% efficient engine.

PROJECTED FUEL COSTS



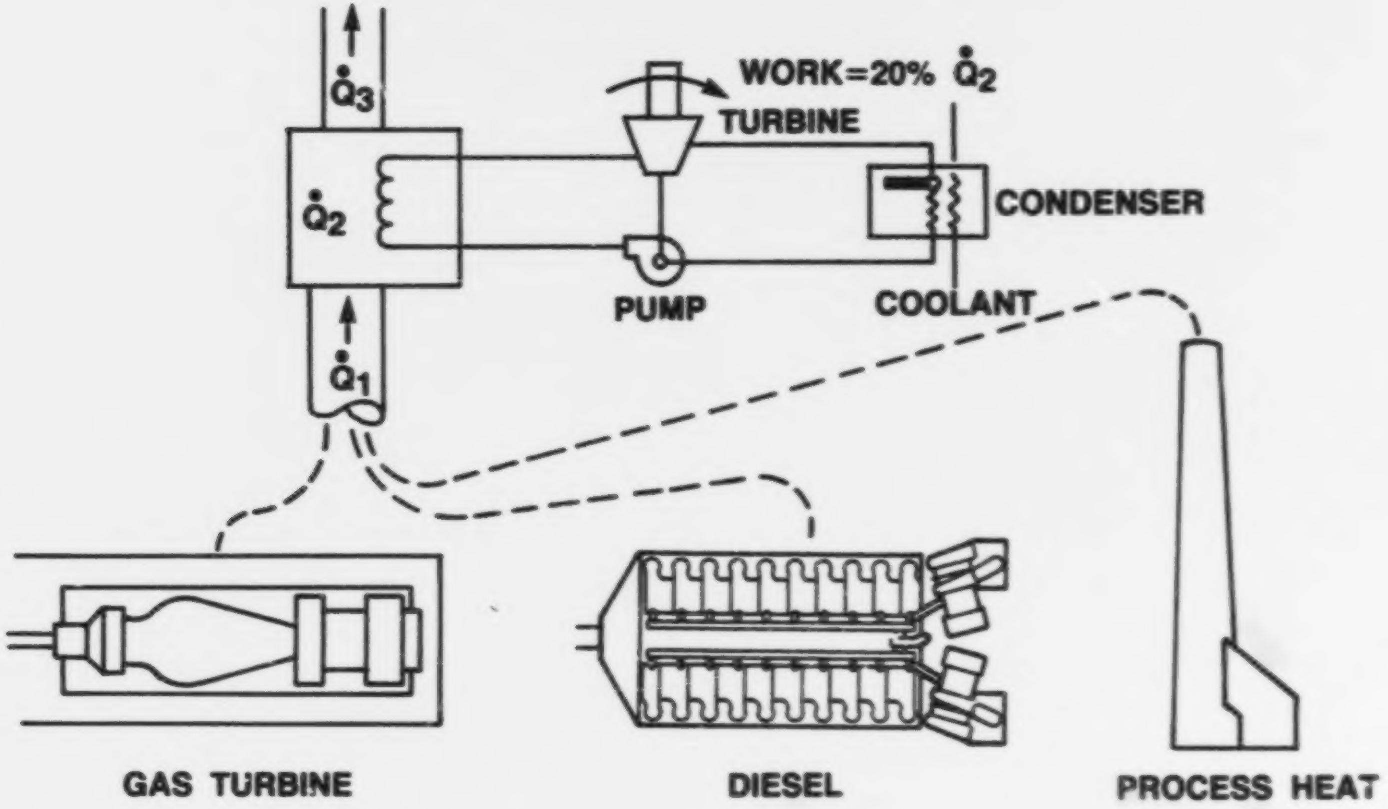
FORD, BACON, & DAVIS INC.—1975
ENERGY MGMT. GUIDEBOOK

AVERAGE INDUSTRIAL RATES PER KWH



SOURCE: FEDERAL POWER COMMISSION, TYPICAL POWER BILLS 1975 FPC R86

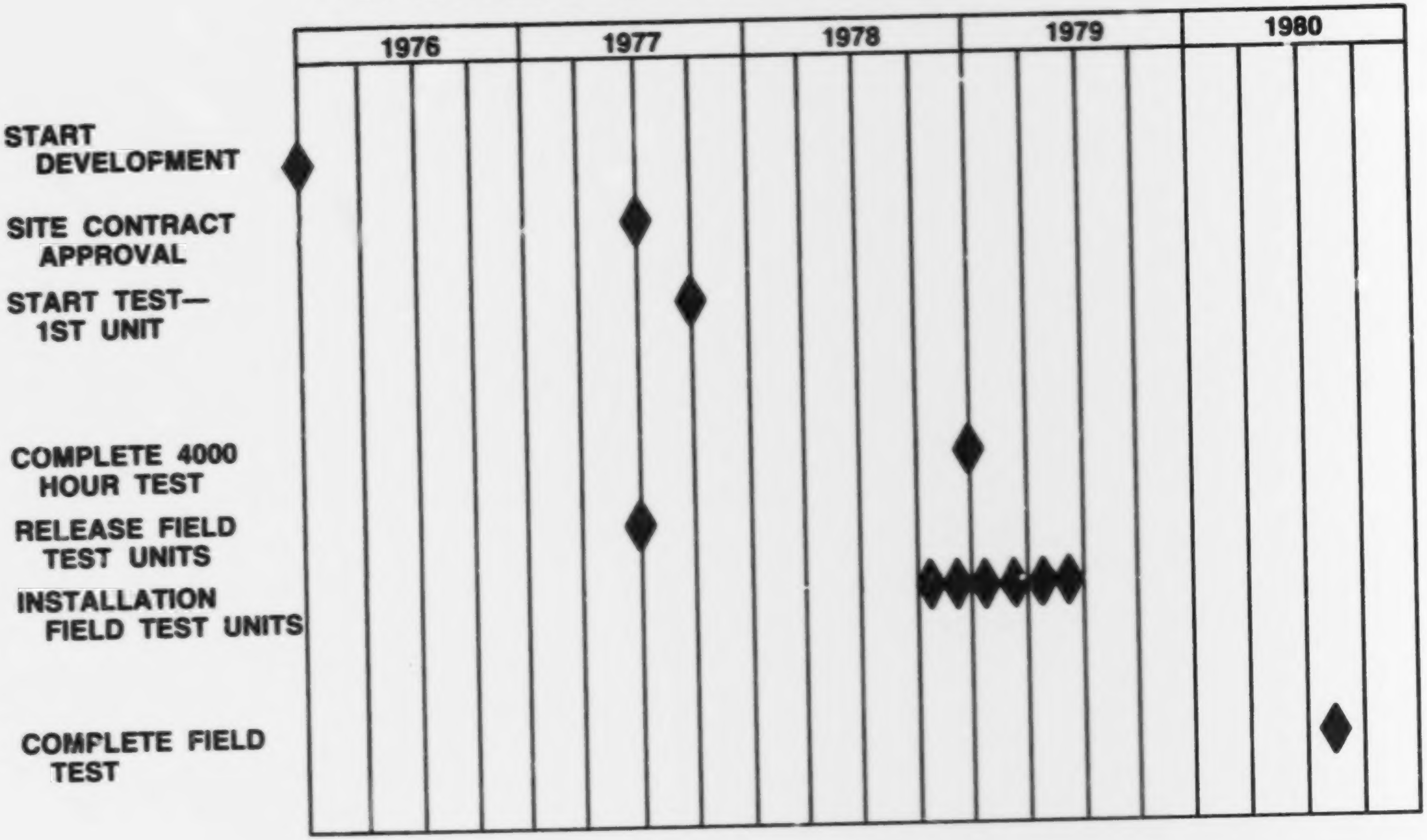
BOTTOMING CYCLE


$\dot{Q}_3$
WORK=20% $\dot{Q}_2$
TURBINE
$\dot{Q}_2$
CONDENSER
PUMP
COOLANT
$\dot{Q}_1$
GAS TURBINE
DIESEL
PROCESS HEAT

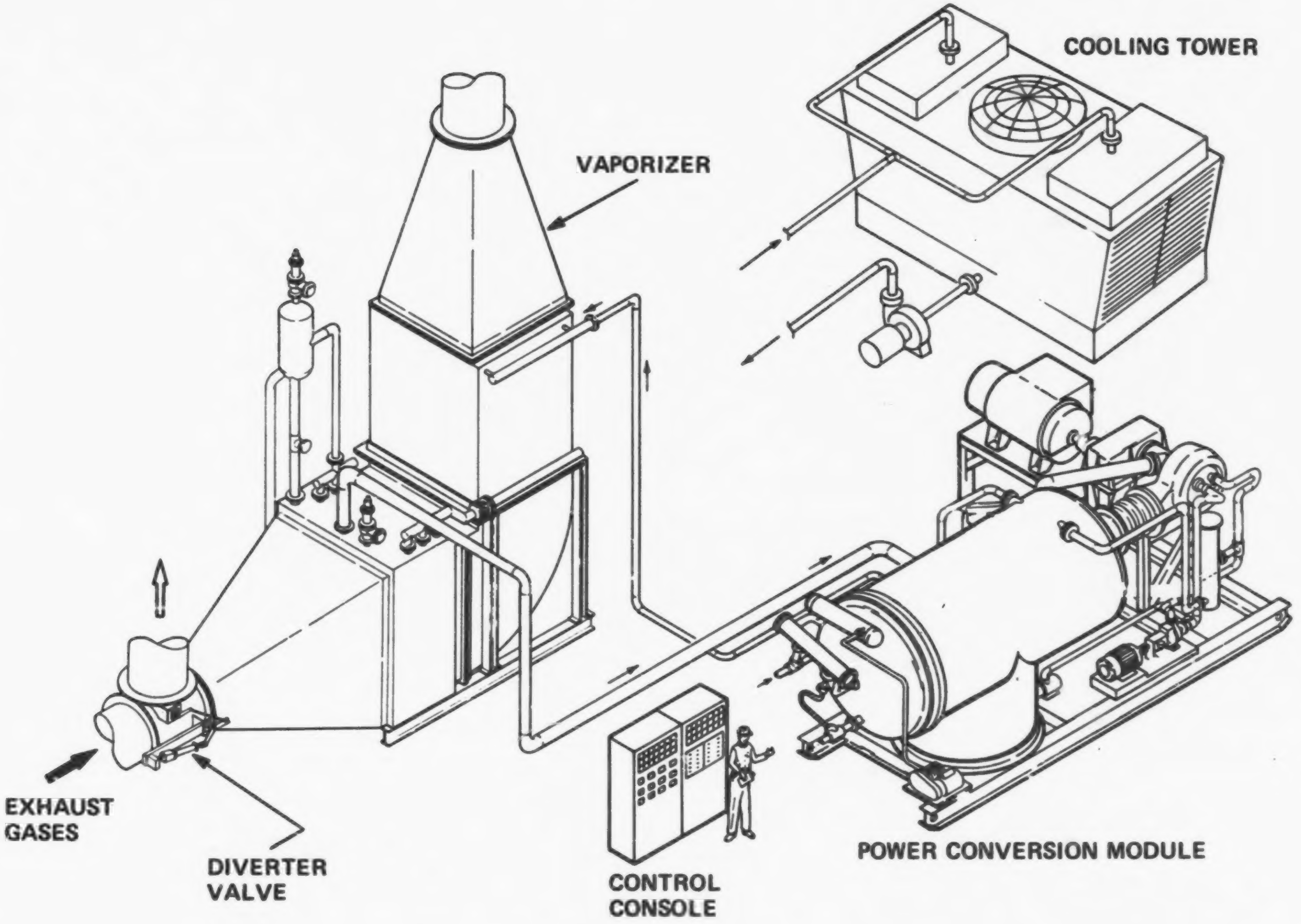
600 KW
ORC System Parameters

- NET OUTPUT: 900 SHP
613 KW
- CYCLE EFFICIENCY: .205 for 200 psi
.227 for 300 psi
- HEAT SINK: 86°F WATER
- WORKING FLUID: TOLUENE

PROGRAM SCHEDULE

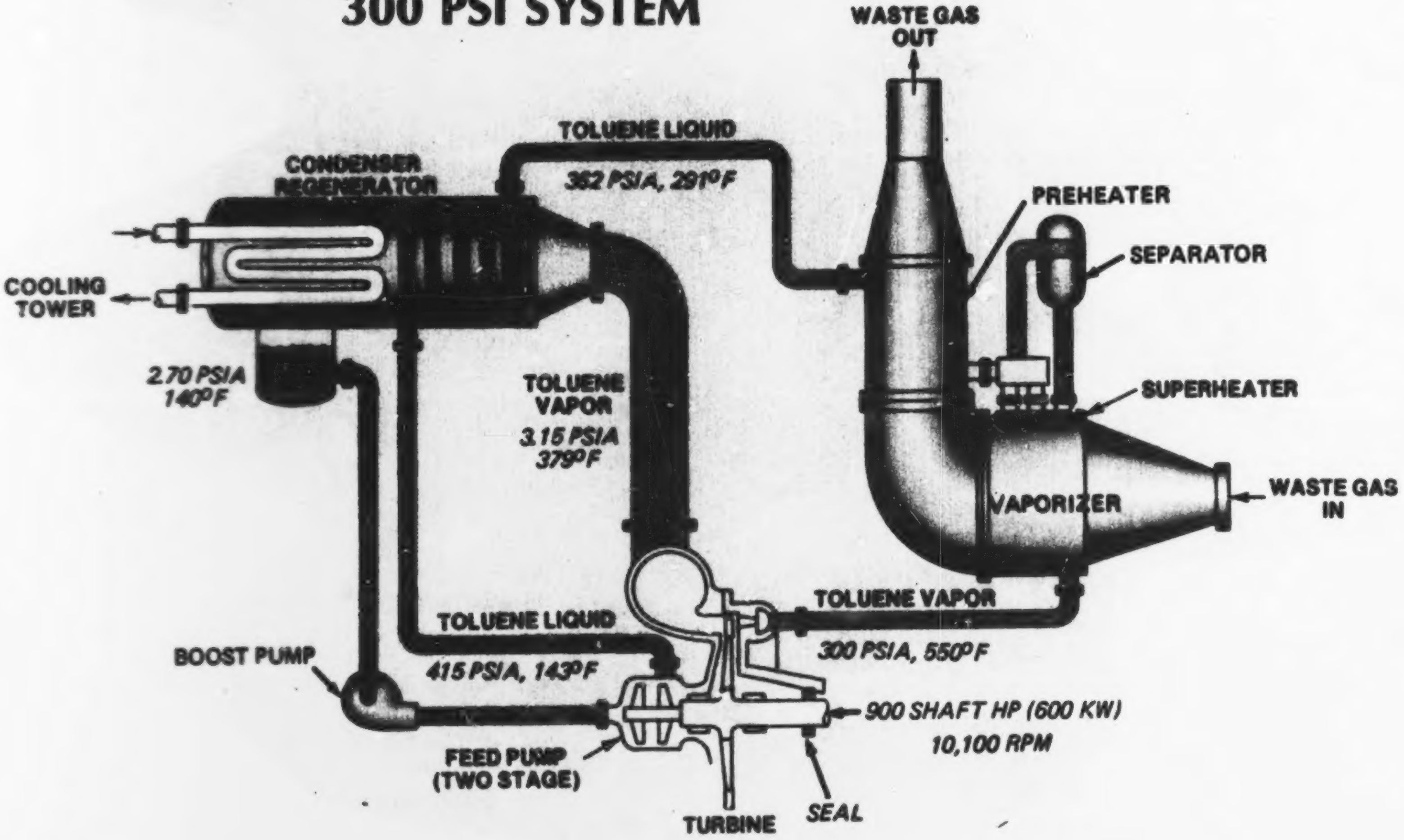


600 KW ORC BOTTOMING SYSTEM


COOLING TOWER
VAPORIZER
EXHAUST GASES
DIVERTER VALVE
CONTROL CONSOLE
POWER CONVERSION MODULE

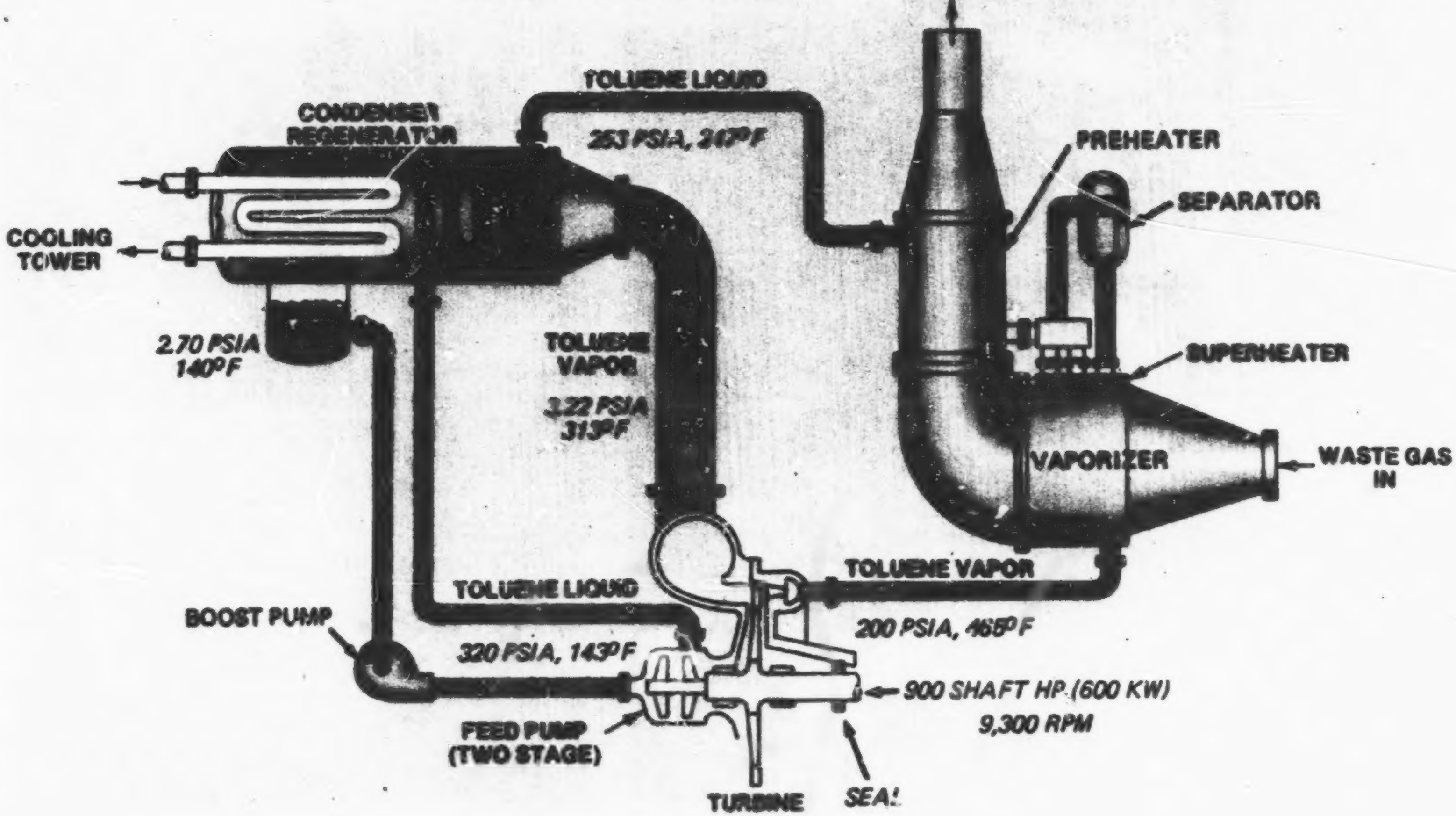
600 KW ORC SCHEMATIC

300 PSI SYSTEM

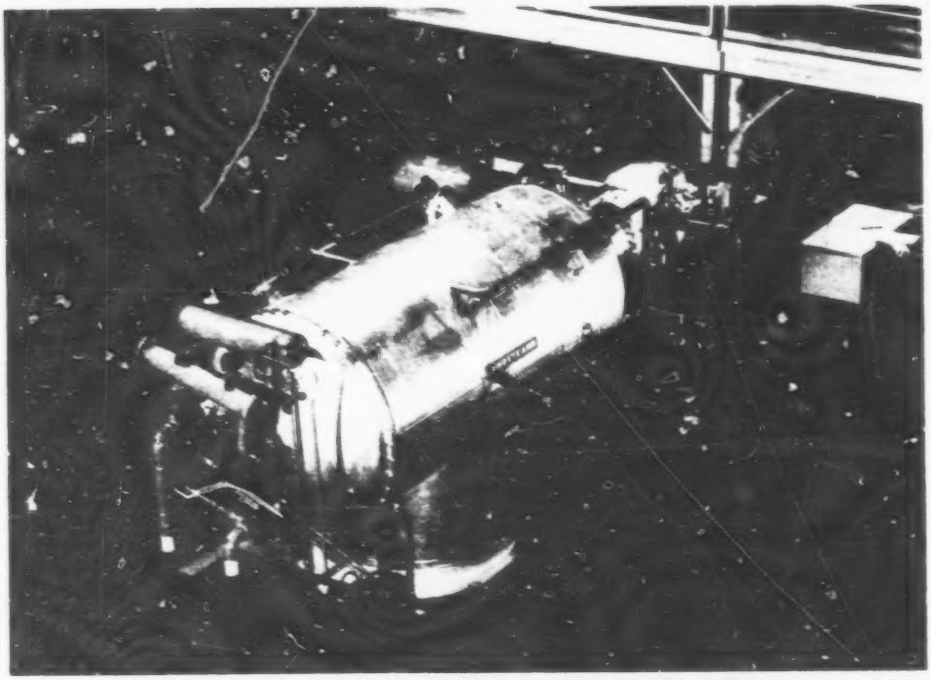

WASTE GAS OUT
TOLUENE LIQUID
362 PSIA, 291°F
CONDENSER REGENERATOR
PREHEATER
SEPARATOR
COOLING TOWER
2.70 PSIA 140°F
TOLUENE VAPOR
3.15 PSIA 379°F
SUPERHEATER
VAPORIZER
WASTE GAS IN
TOLUENE VAPOR
300 PSIA, 550°F
TOLUENE LIQUID
415 PSIA, 143°F
BOOST PUMP
900 SHAFT HP (600 KW)
10,100 RPM
FEED PUMP (TWO STAGE)
TURBINE
SEAL

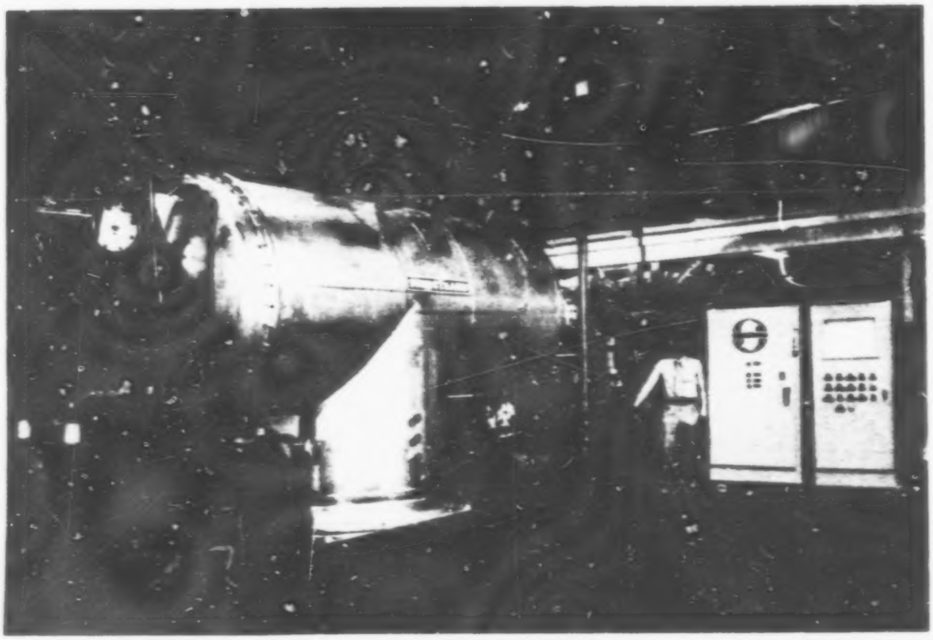
600 KW ORC SCHEMATIC

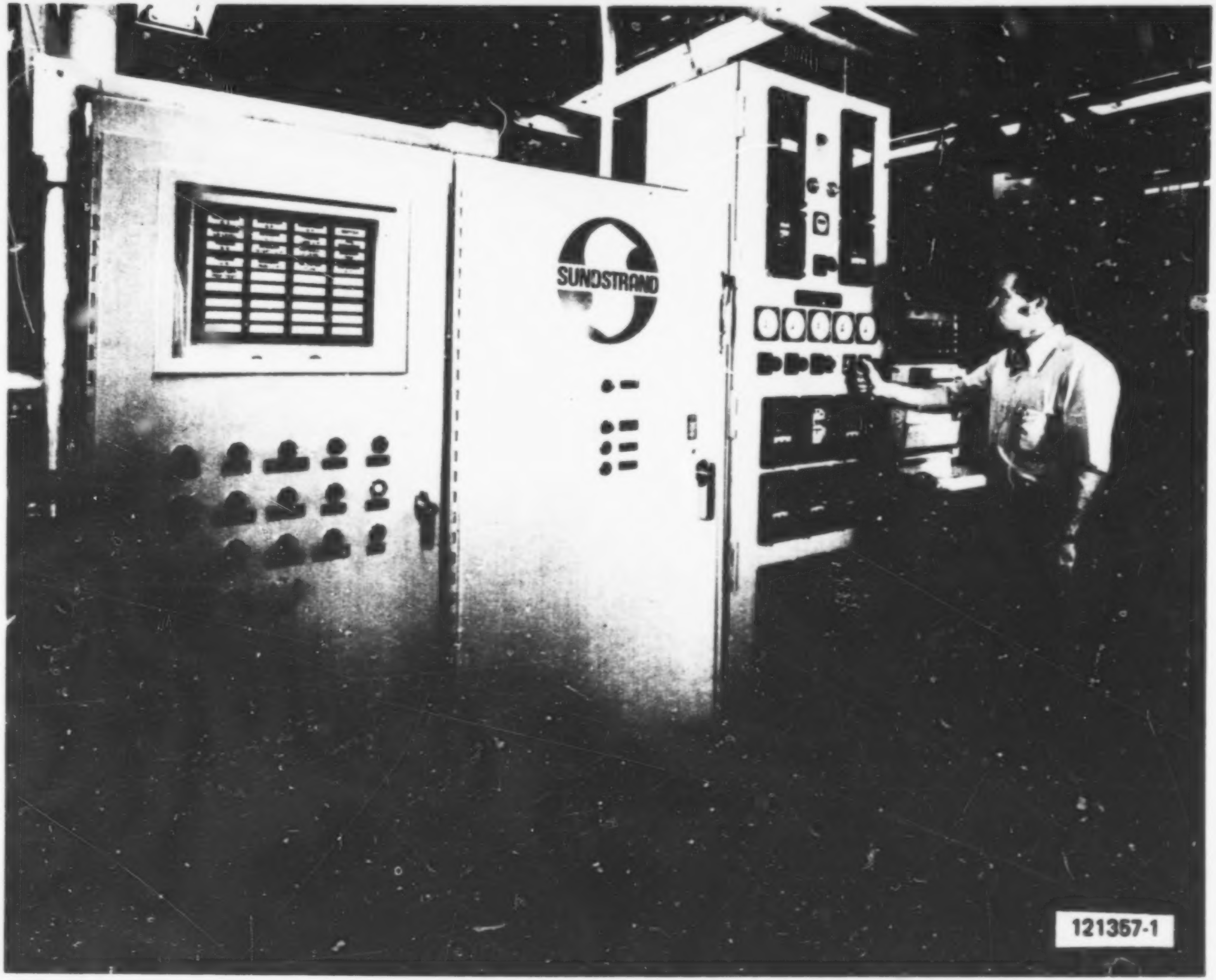
200 PSI SYSTEM


WASTE GAS OUT
TOLUENE LIQUID
CONDENSER REGENERATOR
253 PSIA, 247°F
PREHEATER
SEPARATOR
COOLING TOWER
2.70 PSIA 140°F
TOLUENE VAPOR
3.22 PSIA 313°F
SUPERHEATER
VAPORIZER
WASTE GAS IN
TOLUENE VAPOR
TOLUENE LIQUID
200 PSIA, 465°F
BOOST PUMP
320 PSIA, 143°F
900 SHAFT HP (600 KW)
9,300 RPM
FEED PUMP (TWO STAGE)
TURBINE
SEAL

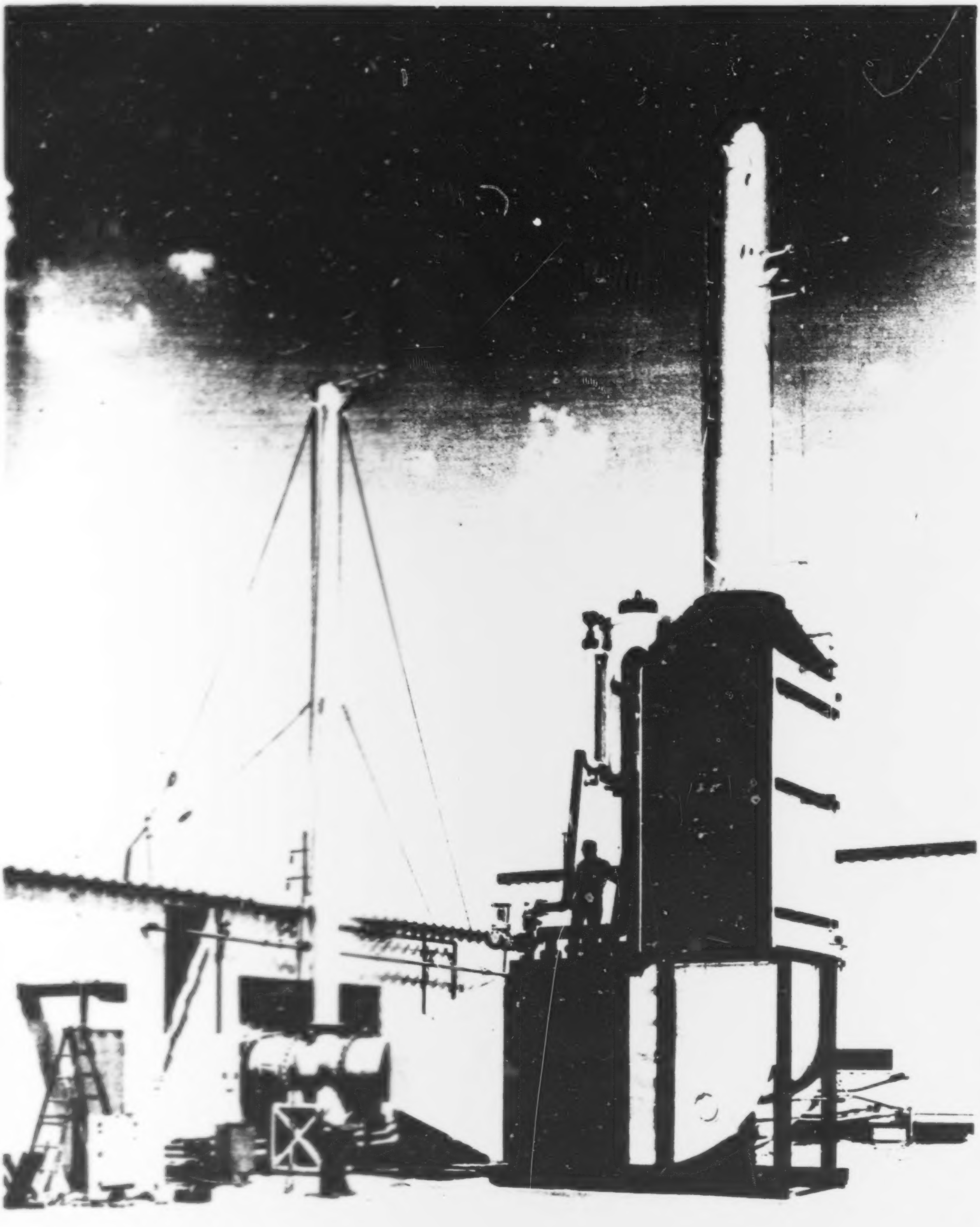
Power Conversion Module



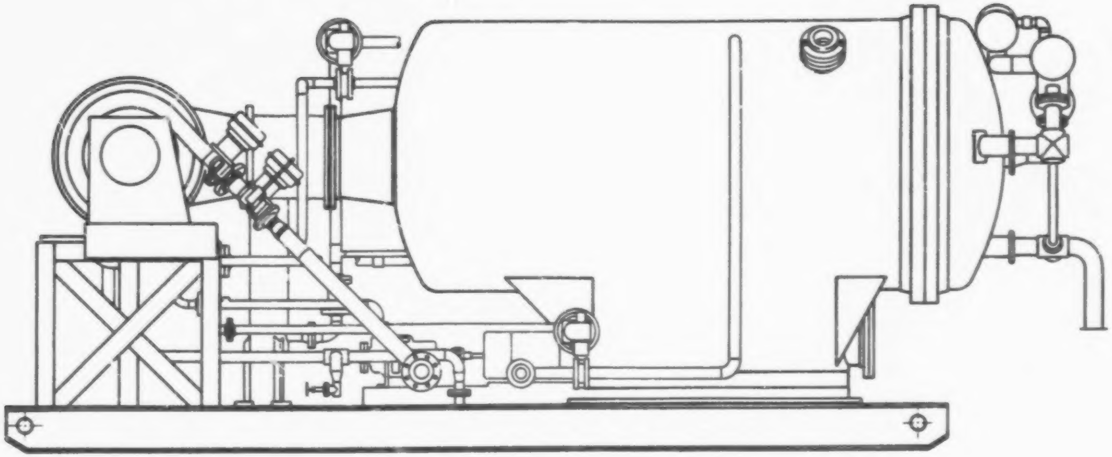



SUNDSTRAND

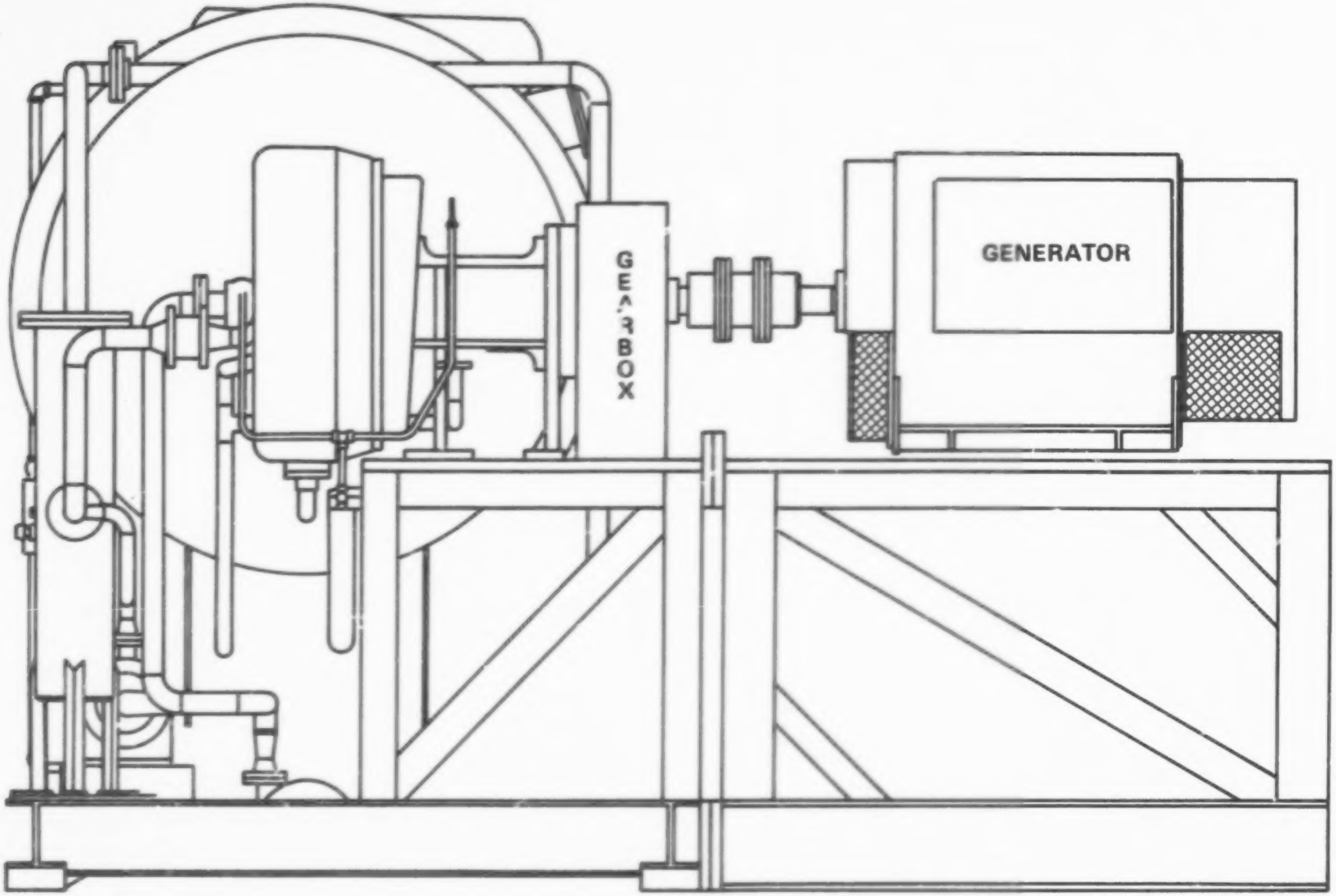
121357-1

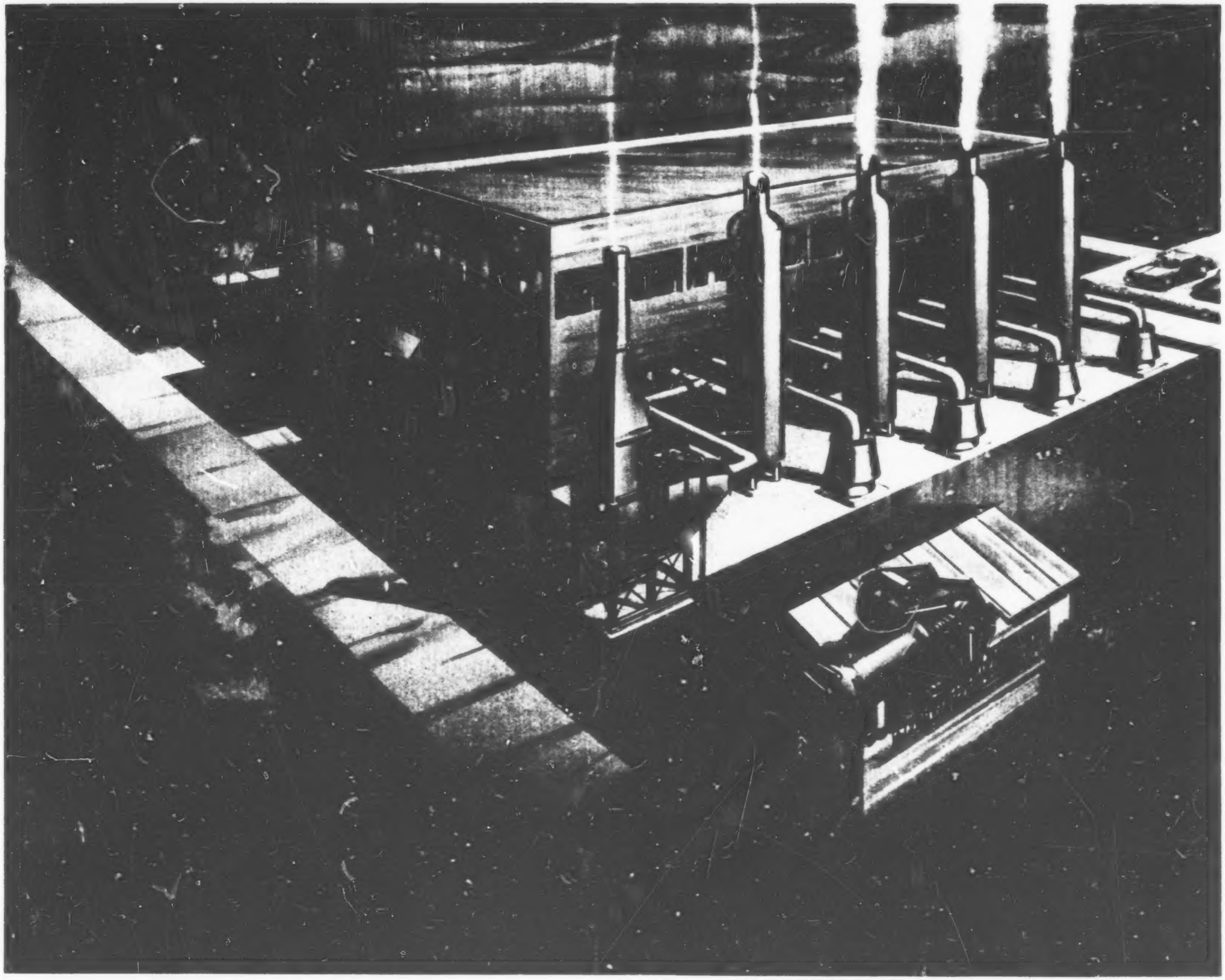


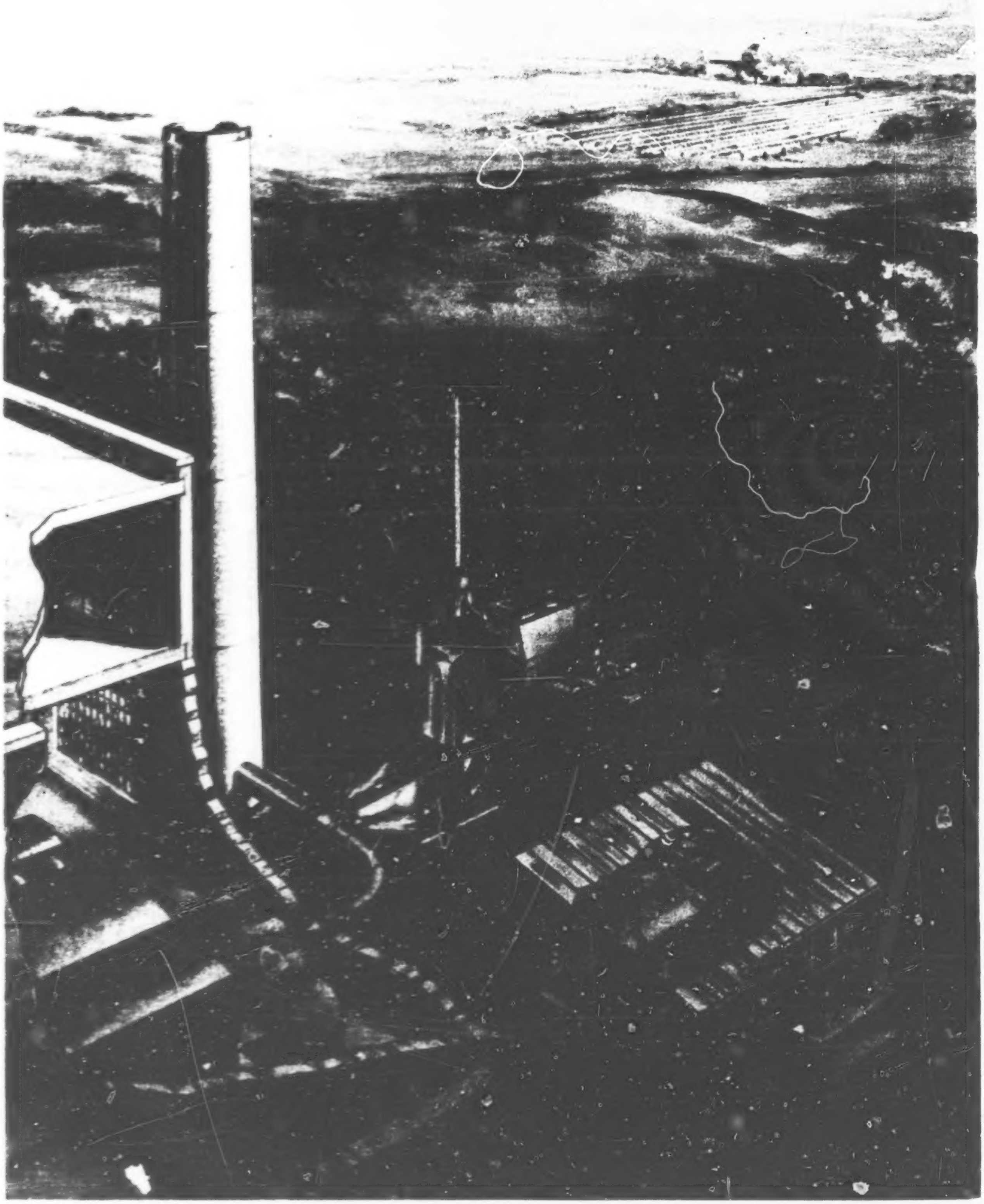
ORC600



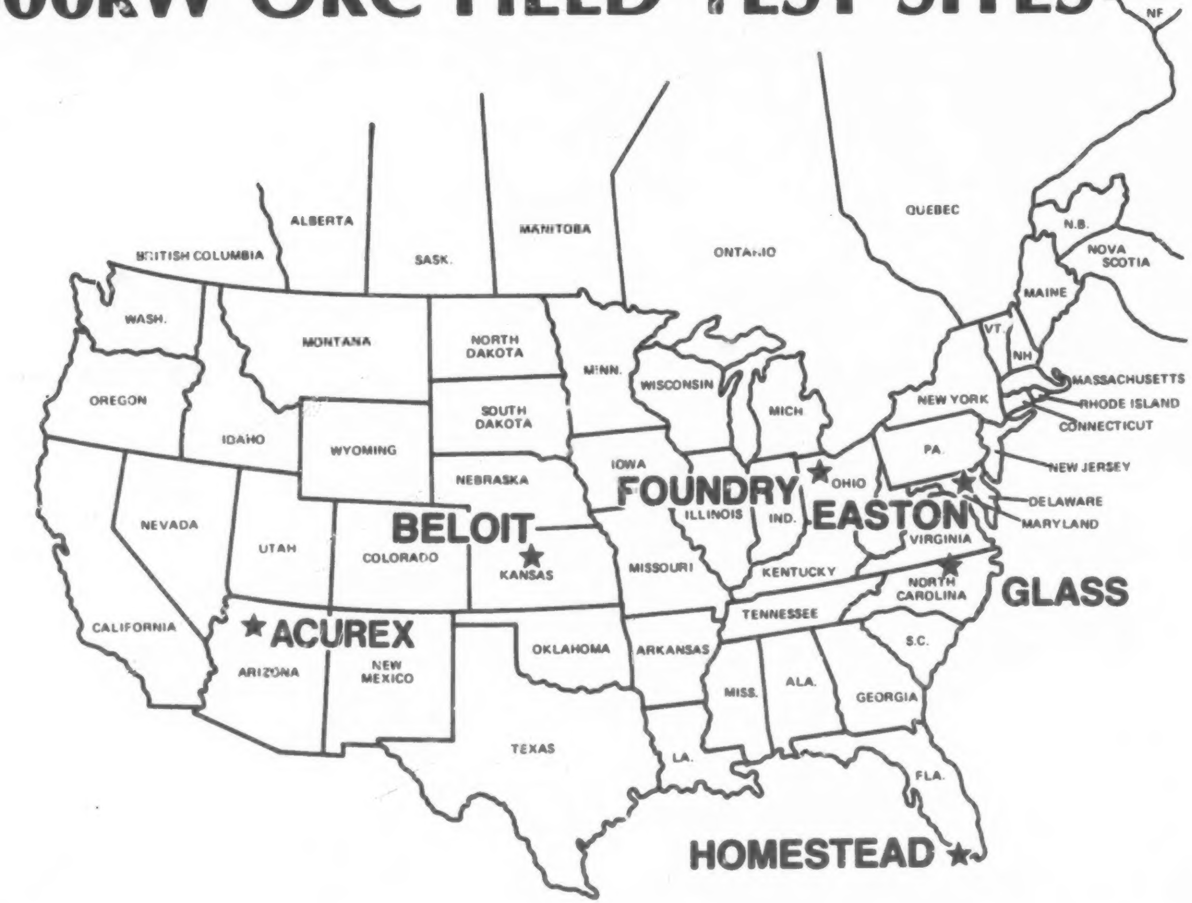
ORC600


GEARBOX
GENERATOR





600kW ORC FIELD TEST SITES


NF
BRITISH COLUMBIA
ALBERTA
SASK.
MANITOBA
ONTARIO
QUEBEC
N.B.
NOVA SCOTIA
WASH.
MONTANA
NORTH DAKOTA
MINN.
MAINE
VT.
NH
OREGON
IDAHO
WYOMING
SOUTH DAKOTA
WISCONSIN
MICH
NEW YORK
MASSACHUSETTS
RHODE ISLAND
CONNECTICUT
PA.
NEW JERSEY
DELAWARE
MARYLAND
NEBRASKA
IOWA
FOUNDRY
OHIO
NEVADA
UTAH
COLORADO
BELOIT
KANSAS
ILLINOIS
IND.
EASTON
VIRGINIA
MISSOURI
KENTUCKY
NORTH CAROLINA
GLASS
CALIFORNIA
ACUREX
ARIZONA
NEW MEXICO
OKLAHOMA
ARKANSAS
TENNESSEE
S.C.
MISS.
ALA.
GEORGIA
TEXAS
LA.
FLA.
HOMESTEAD

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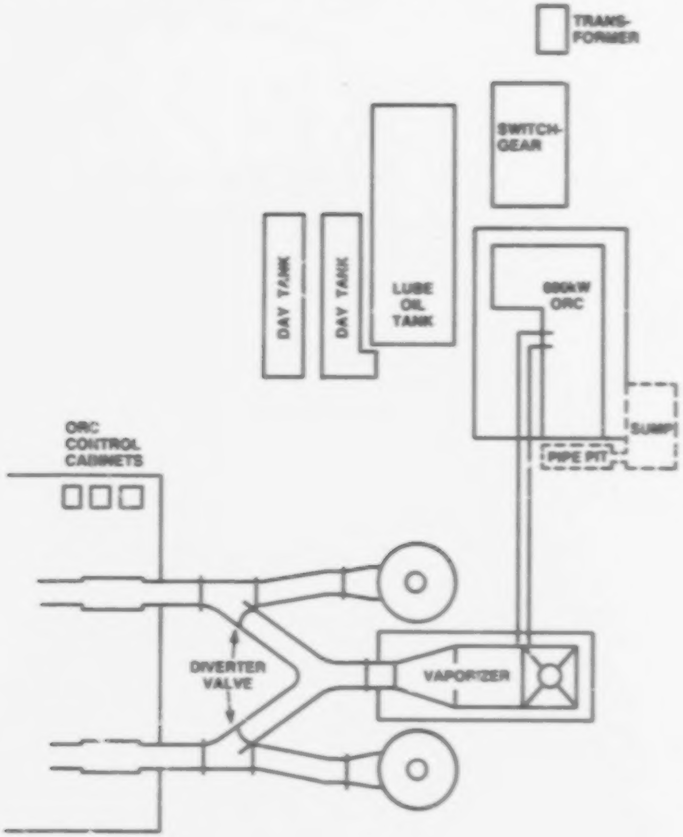
HOMESTEAD

- **HEAT SOURCE**
  - -Engine Make—Delaval
  - -Size (BHP)—12216
  - -Exhaust Temperature—752°F
- **VAPORIZER**
  - -Pressure Drop—7 in. w.c.
  - -Design Flow Rate—128,229 lb/hr.
- **POWER DESIRED**
  - -Electrical—600kW

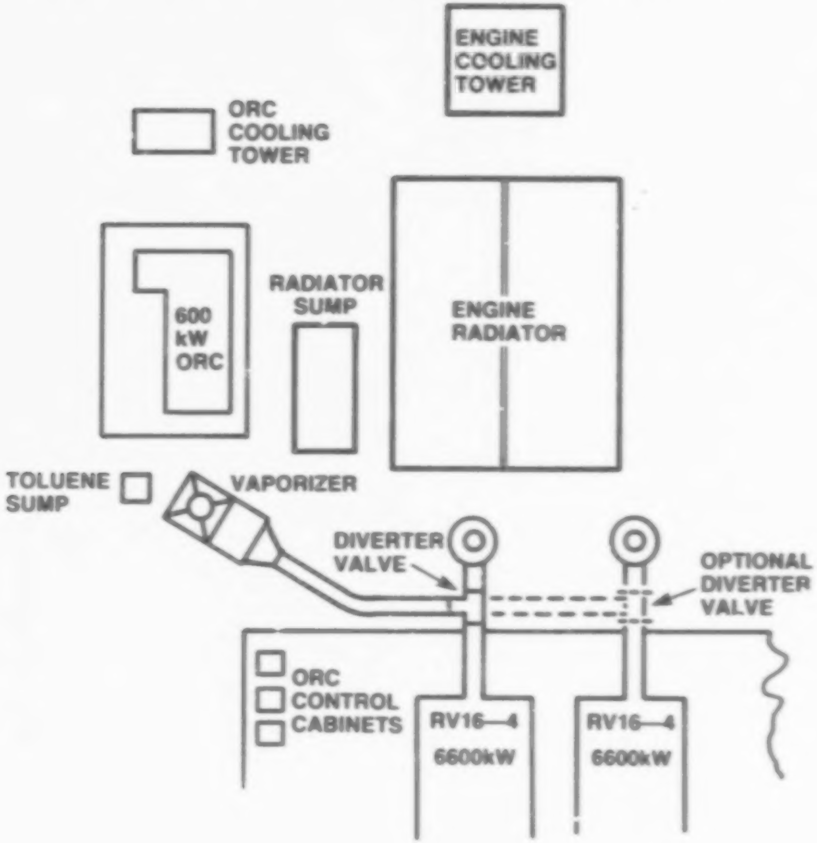
HOMESTEAD, FLORIDA MUNICIPAL UTILITY


TRANS-
FORMER
SWITCH-
GEAR
DAY TANK
DAY TANK
LUBE
OIL
TANK
600kW
ORC
SUMP
PIPE PIT
ORC
CONTROL
CABINETS
DIVERTER
VALVE
VAPORIZER

EASTON

- **HEAT SOURCE**
  - -Engine Make—Delaval
  - -Size (BHP)—-8658
  - -Exhaust Temperature—-752°F
- **VAPORIZER**
  - -Pressure Drop—7 in. w.c.
  - -Design Flow Rate—99,932 lb/hr.
- **POWER DESIRED**
  - -Electrical—600kW

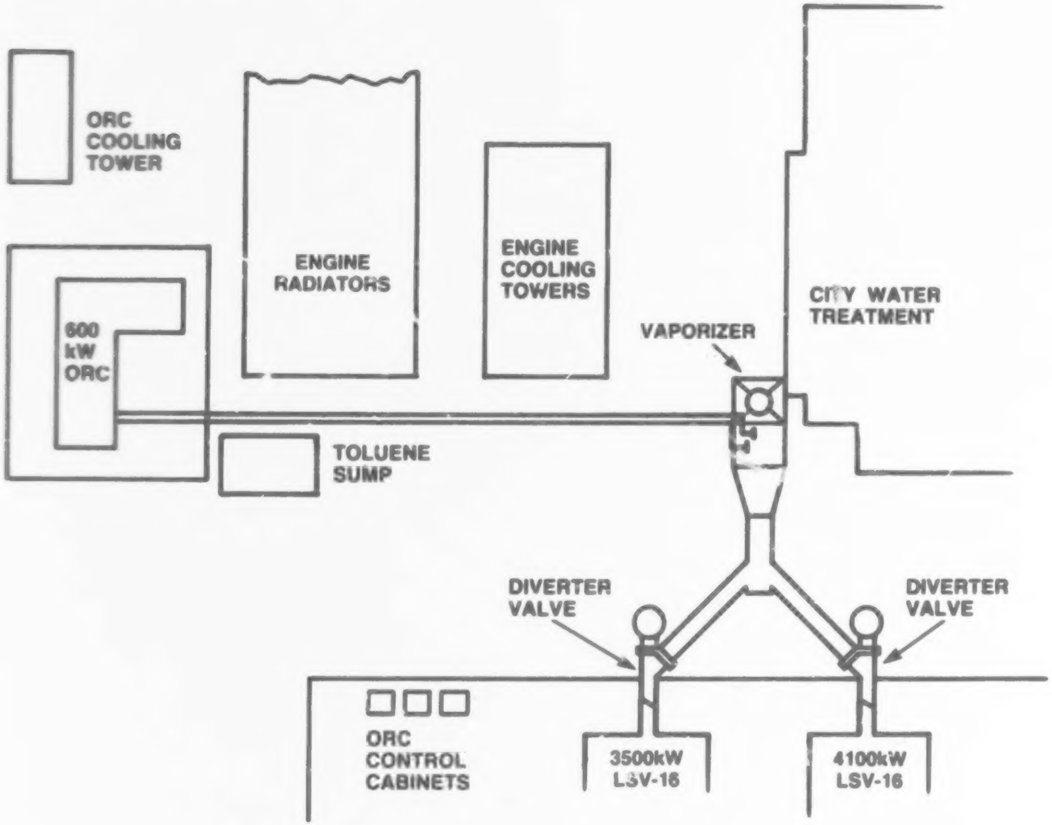
EASTON, MARYLAND
MUNICIPAL UTILITY


ENGINE COOLING TOWER
ORC COOLING TOWER
RADIATOR SUMP
600 kW ORC
ENGINE RADIATOR
TOLUENE SUMP
VAPORIZER
DIVERTER VALVE
OPTIONAL DIVERTER VALVE
ORC CONTROL CABINETS
RV16—4
6600kW
RV16—4
6600kW

BELOIT

- **HEAT SOURCE**
  - -Engine Make—Cooper Bessemer
  - -Size (BHP)—4910 & 5744
  - -Exhaust Temperature—821°F
- **VAPORIZER**
  - -Pressure Drop—7 in. w.c.
  - -Design Flow Rate—94,237 lb/hr.
- **POWER DESIRED**
  - -Electrical—600kW

BELOIT, KANSAS MUNICIPAL UTILITY


ORC COOLING TOWER
ENGINE RADIATORS
ENGINE COOLING TOWERS
CITY WATER TREATMENT
600 kW ORC
VAPORIZER
TOLUENE SUMP
DIVERTER VALVE
DIVERTER VALVE
ORC CONTROL CABINETS
3500kW LSV-16
4100kW LSV-16

FOUNDRY

- **HEAT SOURCE**
  - -Air Preheater
  - -Exhaust Temperature—661°F
- **VAPORIZER**
  - -Pressure Drop—2.5 in. w.c.
  - -Design Flow Rate—150,000 lb/hr.
  - -Draft Fan—26kW
- **POWER DESIRED**
  - -Mechanical—Air Compressor (900 SHP)

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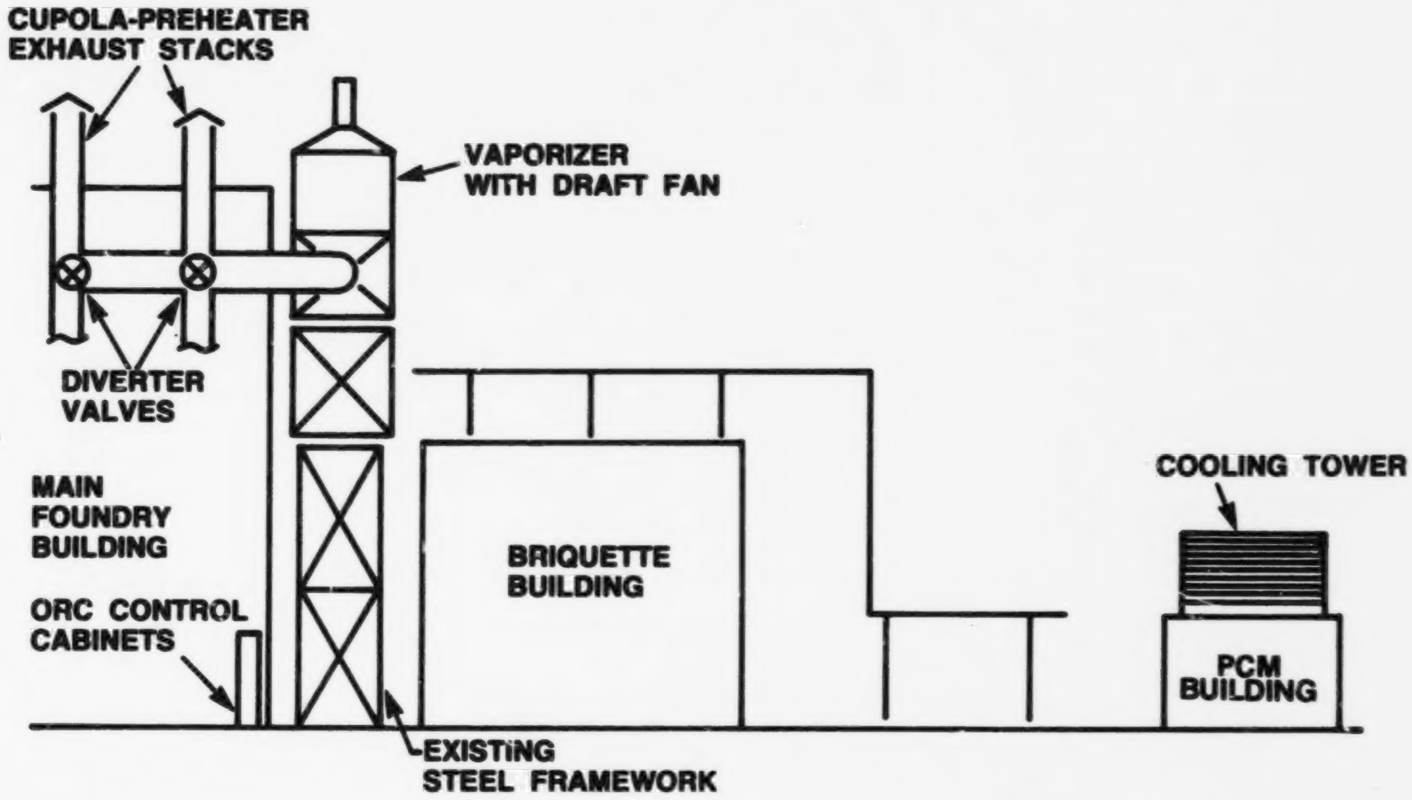
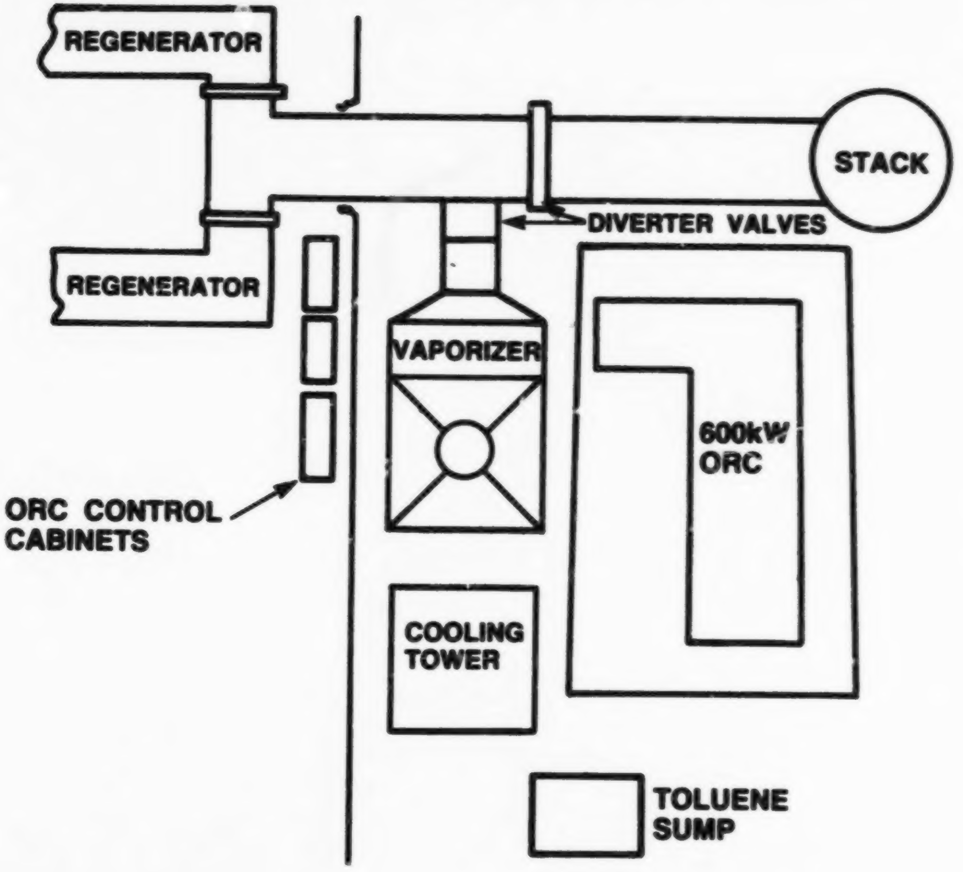
TYPICAL FOUNDRY


CUPOLA-PREHEATER EXHAUST STACKS
VAPORIZER WITH DRAFT FAN
DIVERTER VALVES
MAIN FOUNDRY BUILDING
ORC CONTROL CABINETS
BRIQUETTE BUILDING
COOLING TOWER
PCM BUILDING
EXISTING STEEL FRAMEWORK

GLASS

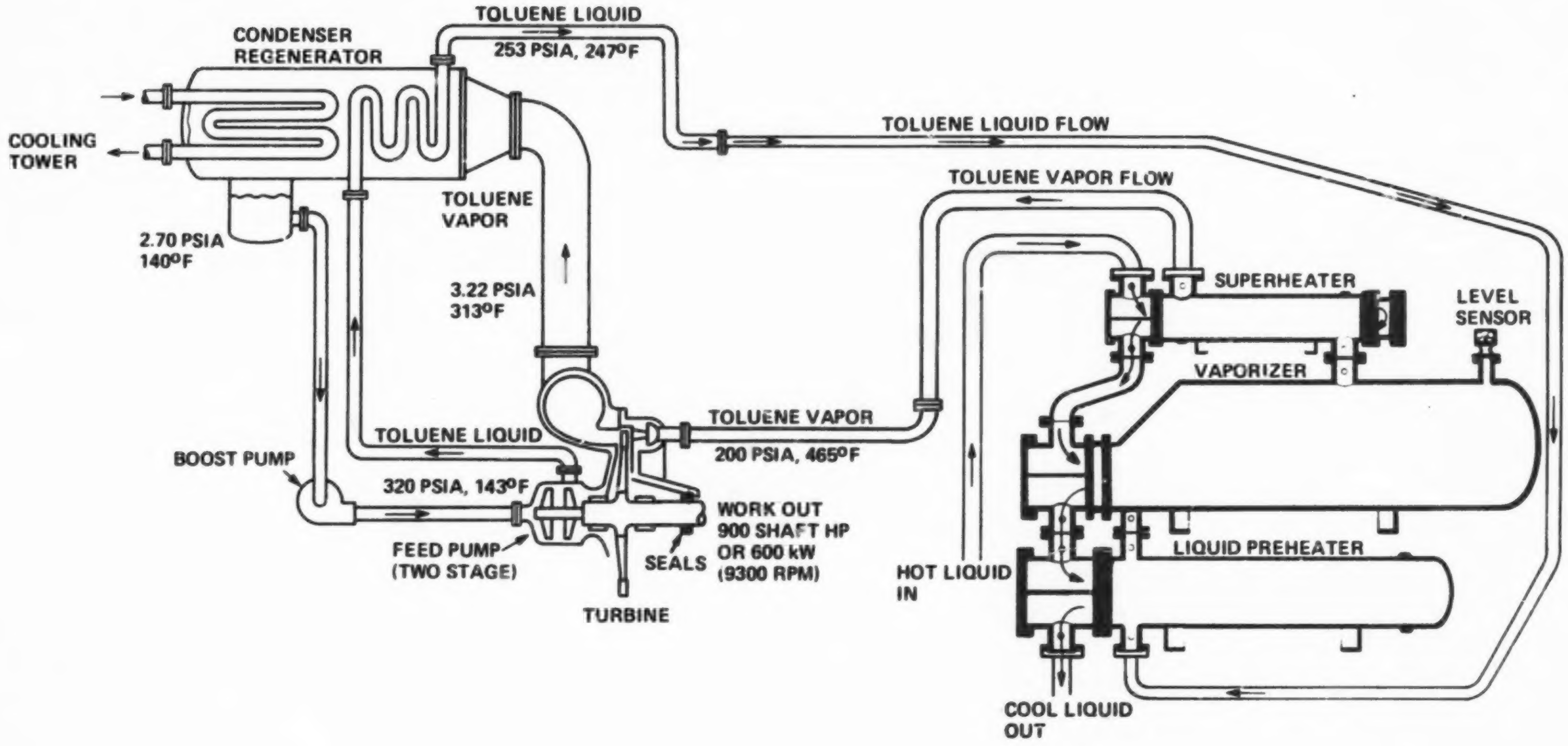
- **HEAT SOURCE**
  - -Type—Glass Tank (2)
  - -Exhaust Temperature—1,600°F
- **VAPORIZER**
  - -Pressure Drop—7 in. w.c.
  - -Design Flow Rate—35,000 lb/hr.
  - -Draft Fan—15kW
- **POWER DESIRED**
  - -Electrical—600kW

TYPICAL GLASS PLANT



REGENERATOR
REGENERATOR
STACK
DIVERTER VALVES
ORC CONTROL CABINETS
VAPORIZER
600kW ORC
COOLING TOWER
TOLUENE SUMP

600kW ORC SCHEMATIC
LIQUID HEAT SOURCE



TOLUENE LIQUID
253 PSIA, 247°F
CONDENSER
REGENERATOR
COOLING
TOWER
TOLUENE LIQUID FLOW
TOLUENE VAPOR FLOW
TOLUENE
VAPOR
2.70 PSIA
140°F
3.22 PSIA
313°F
SUPERHEATER
LEVEL
SENSOR
VAPORIZER
TOLUENE VAPOR
200 PSIA, 465°F
TOLUENE LIQUID
BOOST PUMP
320 PSIA, 143°F
WORK OUT
900 SHAFT HP
OR 600 kW
(9300 RPM)
FEED PUMP
(TWO STAGE)
SEALS
TURBINE
LIQUID PREHEATER
HOT LIQUID
IN
COOL LIQUID
OUT

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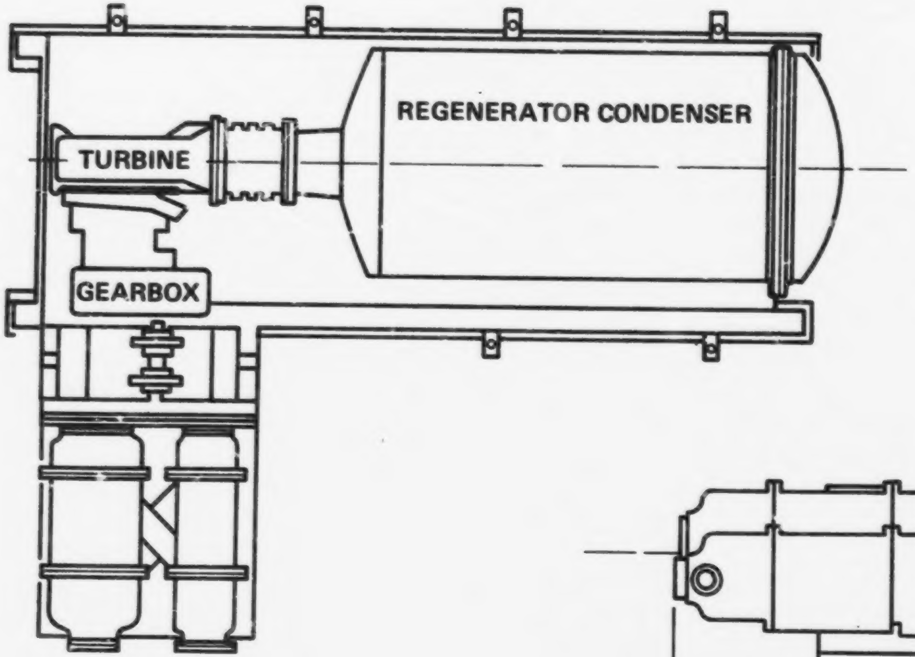
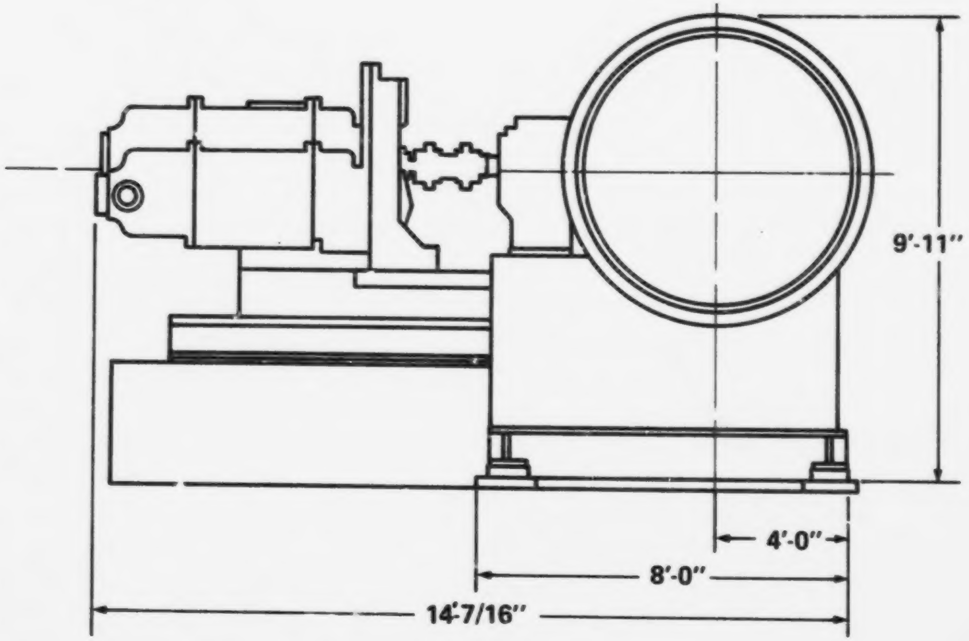
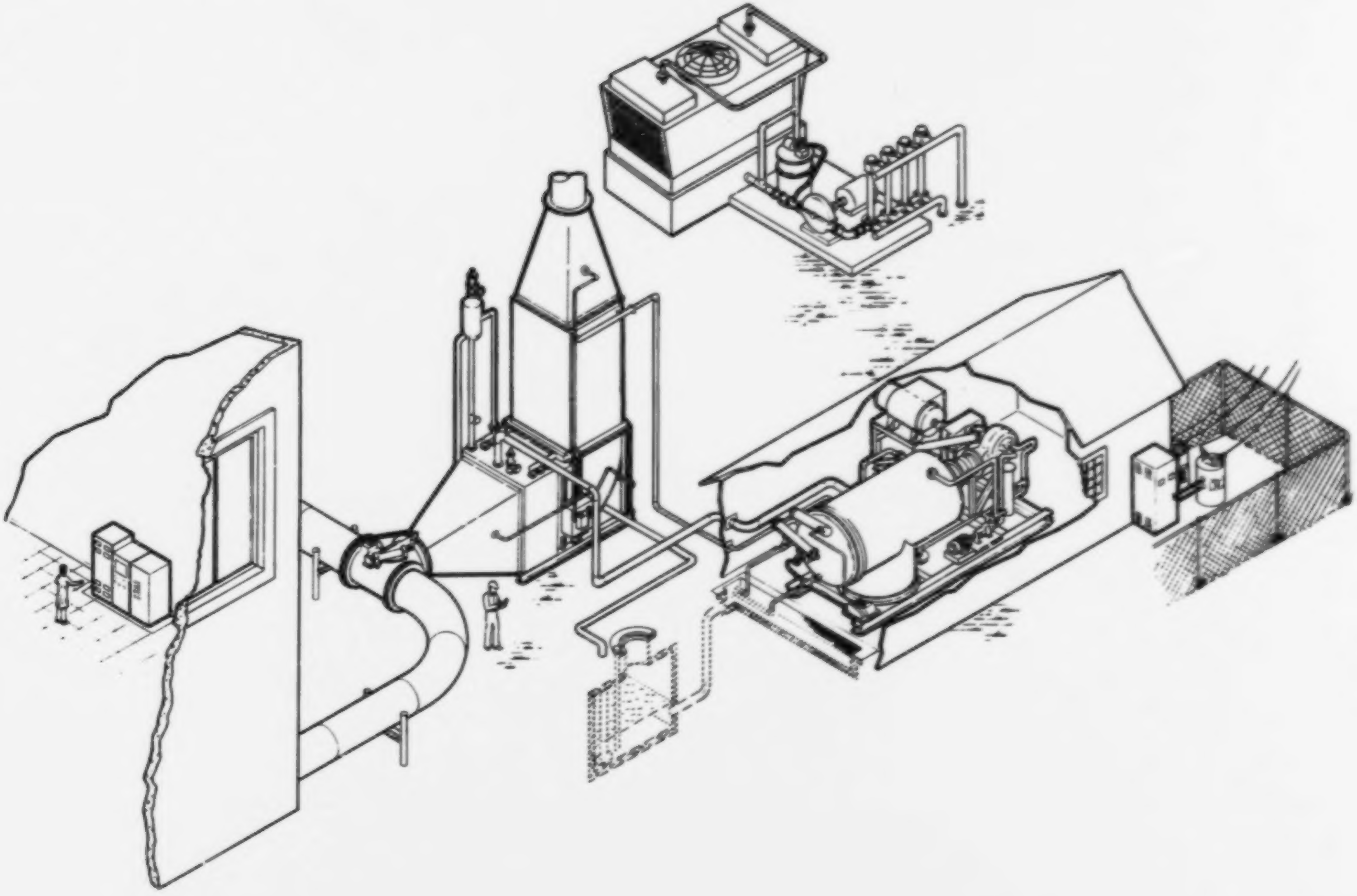
600kW ORC DRIVEN AIR COMPRESSORS


REGENERATOR CONDENSER
TURBINE
GEARBOX


9'-11"
4'-0"
8'-0"
14'7/16"



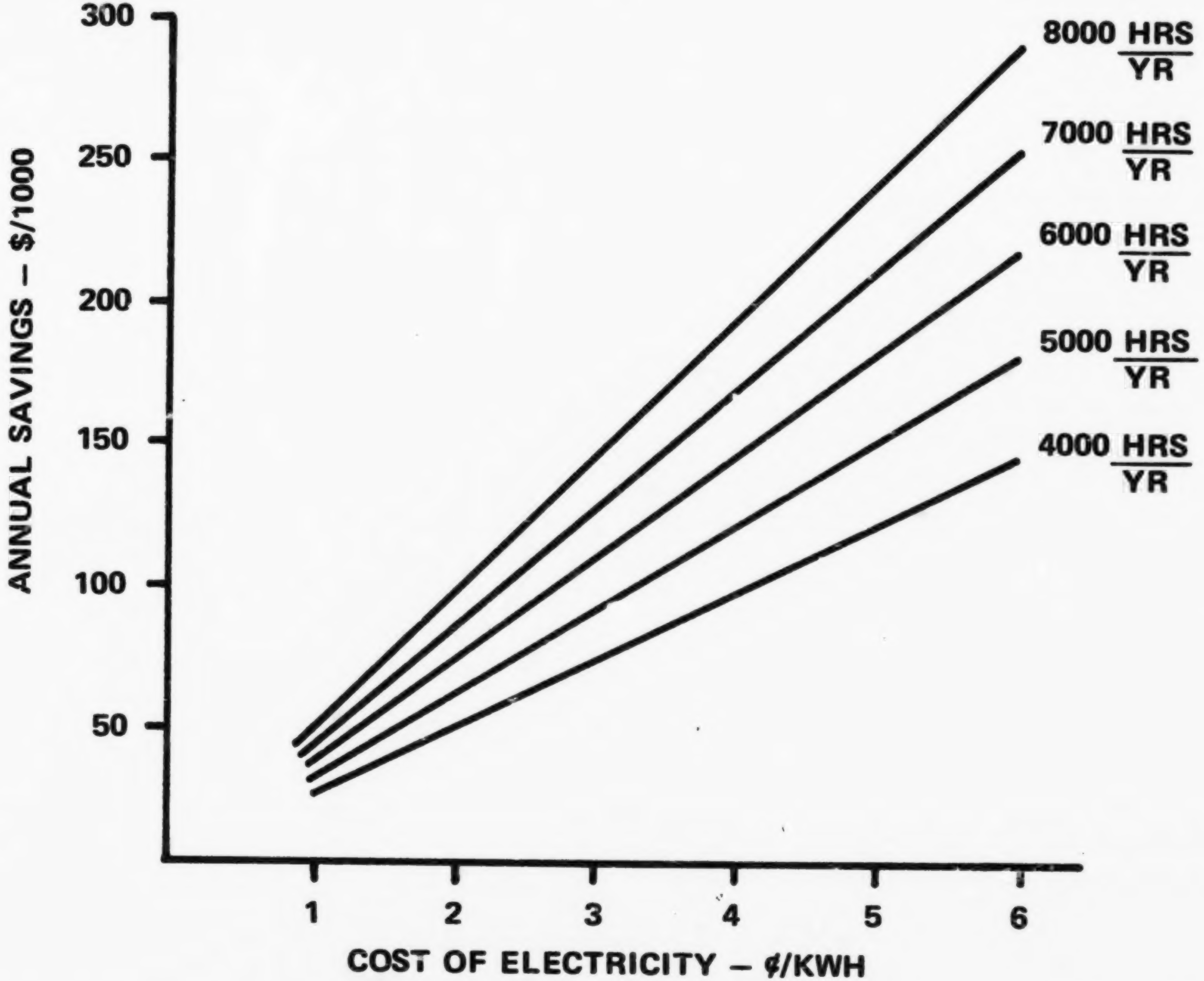
ECONOMICS OF BOTTOMING

- ENERGY CONSERVATION EQUIPMENT
- DOES NOT REDUCE FUEL CONSUMPTION FOR HEAT SOURCE
- COMPETING WITH OTHER OPTIONS
  - Energy
  - $

FACTORS THAT DETERMINE ECONOMICS

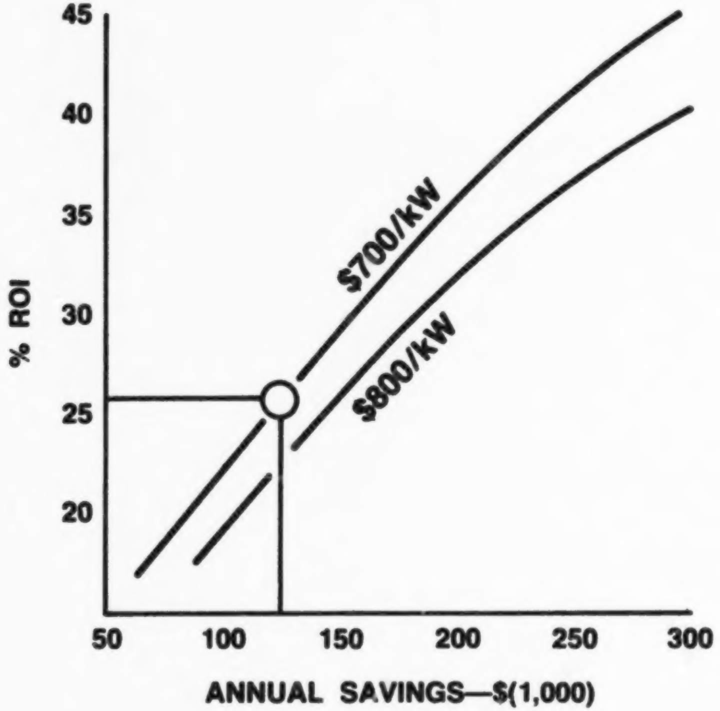
- INSTALLED COST OF BOTTOMING CYCLE
- VALUE OF OUTPUT
- RUNNING HOURS PER YEAR
- OTHER ENERGY CONSERVATION OPTIONS

ANNUAL SAVINGS FROM 600 KW ORC

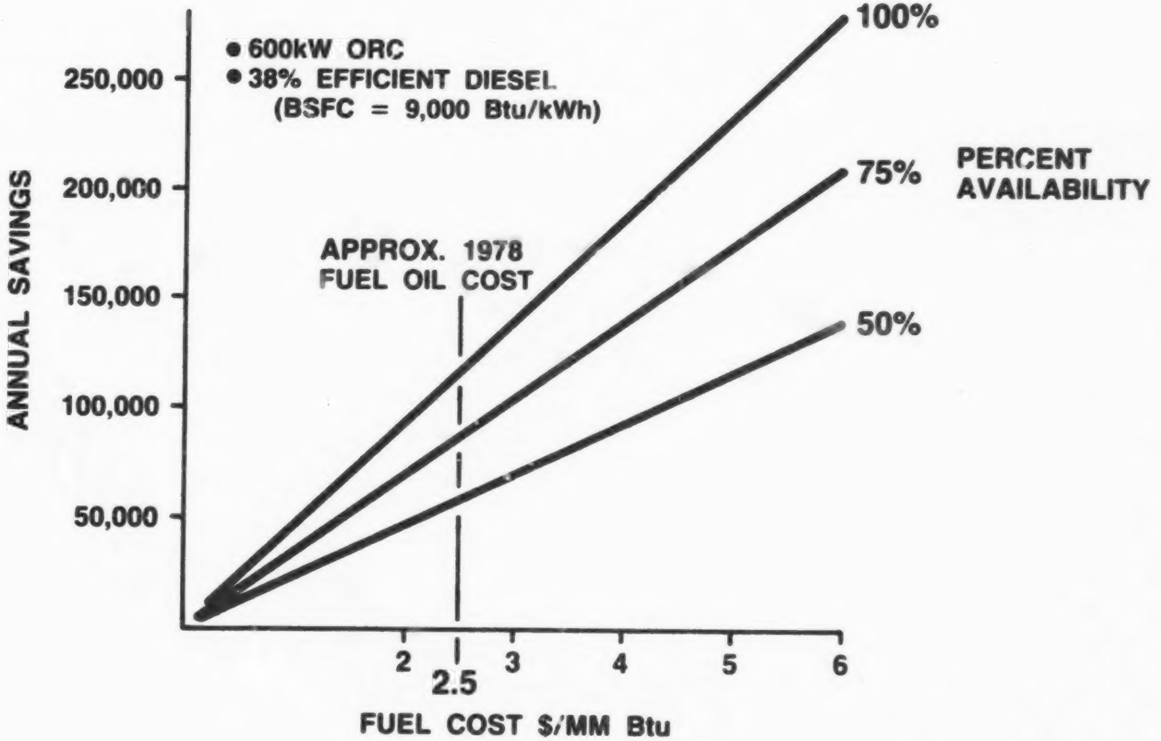


ECONOMICS

Return On Investment and Installed Cost



ANNUAL SAVINGS IN FUEL COST



CHASING BTU'S
OR
SUBATMOSPHERIC GAS TURBINES TO GENERATE POWER

By Robin Mackay
The Garrett Corporation

Presentation to:

Waste Heat Utilization Conference
Sponsored by Engineering Foundation
New England College
Henniker, New Hampshire

16 August 1978

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CHASING BTU'S

OR

SUBATMOSPHERIC GAS TURBINES TO GENERATE POWER

INTRODUCTION

Chasing Btu's is now a highly respectable occupation. Btu's are sought after and accepted almost anywhere they can be found. They lurk in every high temperature stack, whether the stack belongs to a manufacturing process, a fume incinerator, a gas turbine, or a reciprocating engine. The sun is one of the more challenging sources. Fires fueled by low Btu gas, wood chips, agricultural wastes, or other biomass offer good potential. Note that these sources are usually at atmospheric pressure.

In most cases, the energy recovered is passed through a heat exchanger and used simply to produce lower grade heat. Domestic hot water, comfort heating, process steam and process heat are typical outputs. By First Law standards, the efficiency is excellent. Every Btu recovered represents one Btu or more of fuel that does not have to be burned. By Second Law standards, however, the efficiency is poor. Assuming the source heat has a reasonably high temperature, some of those thermal Btu's could be converted to high value mechanical or electrical Btu's while the remainder would still be useful as heat.

This paper will describe two subatmospheric gas turbines currently under development at the AiResearch Manufacturing Company of California which is a division of The Garrett Corporation. Both can generate power using only the energy contained in hot gases at atmospheric pressure.

NOT ALL BTU'S ARE (CREATED) EQUAL

Electrical or mechanical Btu's are far more valuable than thermal Btu's. Oil today typically sells for less than three dollars per million Btu. Three cents per kilowatt hour electricity on the other hand is equivalent to nine dollars per million Btu or more than three times as much. The reason for this is obvious. The average utility-owned central plant is only thirty-two percent efficient at the plant busbar or twenty-nine percent efficient at the customer site if transmission losses are accounted for. It takes a utility almost three and one-half Btu's of thermal energy to deliver one Btu of electrical energy to a customer.

As a simple example, consider a process such as that shown in Figure (1). This could be a metal melting furnace. In the simple process, a burner is installed that burns enough fuel to raise the incoming air temperature by 2340°F (1300°C). With the addition of an 0.82 effective regenerator the incoming air is preheated and the burner need only raise the incoming air temperature by 1159°F (644°C). Assuming a constant c_p for simplicity, fuel consumption is reduced more than 50%. (Higher regenerator effectiveness is possible but the threat of corrosion limits minimum stack temperatures.)

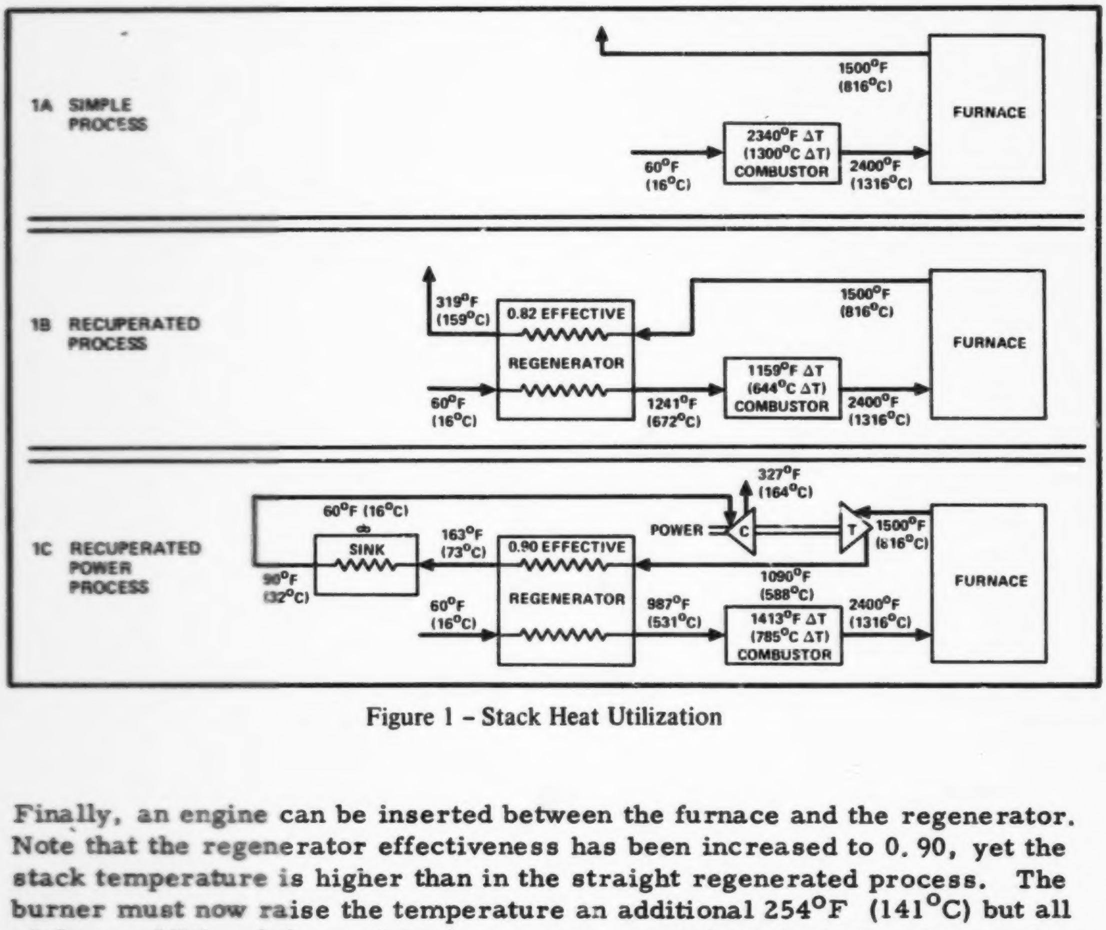

1A SIMPLE PROCESS
1500°F (816°C)
FURNACE
60°F (16°C)
2340°F ΔT (1300°C ΔT) COMBUSTOR
2400°F (1316°C)
1B RECUPERATED PROCESS
319°F (159°C)
0.82 EFFECTIVE
REGENERATOR
1500°F (816°C)
FURNACE
60°F (16°C)
1241°F (672°C)
1159°F ΔT (644°C ΔT) COMBUSTOR
2400°F (1316°C)
1C RECUPERATED POWER PROCESS
327°F (164°C)
60°F (16°C)
POWER
C
T
1500°F (816°C)
SINK
163°F (73°C)
0.90 EFFECTIVE
90°F (32°C)
REGENERATOR
1090°F (588°C)
FURNACE
60°F (16°C)
987°F (531°C)
1413°F ΔT (785°C ΔT) COMBUSTOR
2400°F (1316°C)

Figure 1 – Stack Heat Utilization

Finally, an engine can be inserted between the furnace and the regenerator. Note that the regenerator effectiveness has been increased to 0.90, yet the stack temperature is higher than in the straight regenerated process. The burner must now raise the temperature an additional 254°F (141°C) but all of these additional thermal Btu's are compensated for by high value mechanical Btu's produced by the engine. Indeed, assuming an 85% efficient compressor and a 90% efficient turbine, the engine will put out 54 Btu per lb of air flow not counting parasitic or radiation losses. Yet the increased fuel required to produce the additional 254°F (141°C) is less than 68 Btu per pound of air flow. This is the equivalent of producing power at about 80% efficiency.

In the simple process, enough fuel is purchased to heat the incoming air by 2340°F (1300°C) and the process functions. In the regenerated process, only enough fuel is bought to heat the incoming air by 1159°F (644°C) but the process still functions. In the regenerated power process, enough fuel is bought to heat the incoming air 1413°F (785°C), but the power produced as a byproduct is worth more than twice as much as the incremental fuel bill and the process still functions!

SUBATMOSPHERIC GAS TURBINES

Before trying to insert hard numbers into an actual process, we should examine the type of engine that will operate on a 1500°F (816°C) to 1650°F (899°C) inlet temperature and still provide a source of hot gases in the 1000°F (538°C) to 1200°F (649°C) range. It could well be an open cycle gas turbine with the combustor replaced by a heat exchanger. Thermodynamically, the concept is sound. From a practical standpoint, the construction of a heat exchanger to cope simultaneously with pressures of several atmospheres and temperatures of 1500°F (816°C) at reasonable cost is extremely difficult. Accordingly we look to the subatmospheric gas turbine as a solution which eliminates the requirement for heat transfer with metal temperatures over 1200°F (649°C).

The components of a subatmospheric gas turbine are similar to those of a conventional regenerated turbine. It is just that the flow path is slightly different. Consider a basic simple cycle low pressure ratio gas turbine as shown in Figure (2). Air is compressed and the temperature rises from an

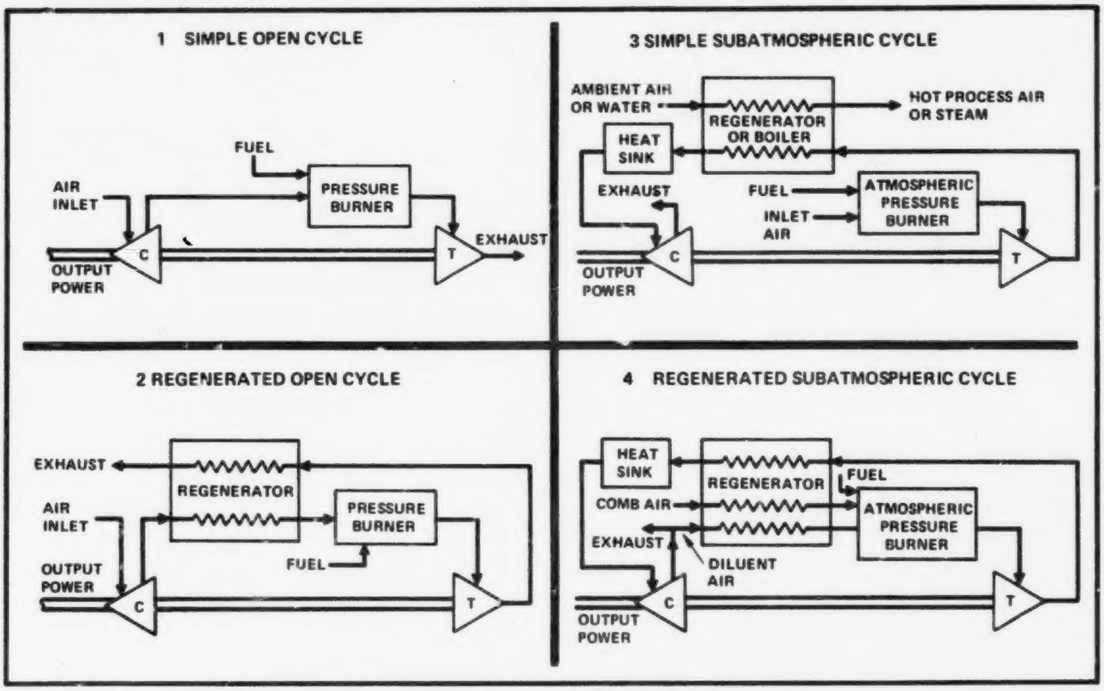

1 SIMPLE OPEN CYCLE
FUEL
AIR INLET
PRESSURE BURNER
EXHAUST
C
T
OUTPUT POWER
3 SIMPLE SUBATMOSPHERIC CYCLE
AMBIENT AIR OR WATER
REGENERATOR OR BOILER
HOT PROCESS AIR OR STEAM
HEAT SINK
EXHAUST
FUEL
INLET AIR
ATMOSPHERIC PRESSURE BURNER
C
T
OUTPUT POWER
2 REGENERATED OPEN CYCLE
EXHAUST
REGENERATOR
AIR INLET
PRESSURE BURNER
FUEL
OUTPUT POWER
C
T
4 REGENERATED SUBATMOSPHERIC CYCLE
HEAT SINK
REGENERATOR
FUEL
COMB AIR
ATMOSPHERIC PRESSURE BURNER
EXHAUST
DILUENT AIR
C
T
OUTPUT POWER

Figure 2 - Gas Turbine Cycle

initial 80°F (27°C) to perhaps 380°F (19[illegible]°C). Sufficient heat is added (usually in a combustor) to bring the temperature up to about 1600°F (871°C). The hot air expands through the turbine, producing sufficient power to drive the compressor, the accessories, any parasitic losses and, of course, the load. The air then discharges at typically 1115°F (602°C).

The addition of a regenerator transfers heat from the exhaust to the compressor discharge air, thus the combustor receives air in the 1040°F (560°C) range. The combustor, therefore, need raise the temperature only 660°F (367°C) rather than the 1220°F (678°C) in the simple cycle. Fuel consumption is reduced by more than half and the efficiency is increased accordingly.

The problems start as we substitute an external source of heat for the combustor. Combustion technology is capable of producing temperatures well beyond what a turbine nozzle and wheel can take. However, if the heat must be transferred into the cycle through a heat exchanger, the turbine inlet temperature is limited by the maximum temperature the heat exchanger can take. Keep in mind that the heat exchanger is containing several atmospheres of pressure. With conventional reasonable cost heat exchanger materials, the turbine inlet temperature must drop down into the 1200°F (649°C) range. The loss in specific power and efficiency is dramatic. (Fortunately, the method of reducing turbine inlet temperature in a waste heat application is usually some form of dilution so the mass flow increases, thus partially restoring the absolute power loss.)

The subatmospheric turbine on the other hand relaxes the requirement for a high temperature heat exchanger. The hot gases can flow directly into the turbine wheel because the turbine inlet pressure is slightly below atmospheric. The hot gases expand through the turbine producing power and dropping in temperature. Turbine discharge temperatures are in the 1115°F (602°C) range, cool enough to enter a regenerator made of moderate cost materials. The turbine discharge pressure is, of course, subatmospheric, typically about 4 psia (28 kPa) for a turbine pressure ratio of about 3.7 to 1. The compressor sustains this partial vacuum just as it sustains an elevated pressure in a conventional positive pressure gas turbine.

The atmospheric pressure side of the regenerator is now producing hot air at temperatures in excess of 1000°F (538°C). This high grade heat is available as preheated combustion air or for direct use in a variety of processes.

There has been some concern about the products of combustion condensing out in the regenerator or heat sink. Fortunately, the pressure is low enough in these components to be generally below the water vapor pressure of the combustion products.

The major drawback to a subatmospheric turbine as opposed to a conventional gas turbine is the large size of the components due to the low density of the working fluid. This is traded off against (1) much lower cost heat transfer in an externally combusted engine, and (2) increased component efficiency in low power engines due to increased component size.

TEN-TON HEAT PUMP

The first subatmospheric gas turbine being developed at Garrett is rated at twelve horsepower and drives a ten-ton heat pump. The heat pump side of the system is quite conventional except for two things. First, the Freon-12 compressor is only 1.75" (4.45 cm) in diameter and turns at 80,000 rpm. Second, the Freon compressor is driven through a magnetic coupling so that the refrigeration system is hermetic.

Production versions of this engine are being designed to operate on natural gas. Some versions, however, will be able to operate on hot gases taken directly from an exhaust stack. This unit was originally supported by the American Gas Association. A. G. A. has now turned most of their research over to the Gas Research Institute and G. R. I. is now the sponsor.

An important feature of the subatmospheric cycle is that the combustor operates essentially at atmospheric pressure. Thus, natural gas at normal delivery pressures of a few inches of water can flow into the combustor. In a conventional turbine the gas would have to be compressed to a pressure somewhat above the gas turbine's compressor discharge pressure or the gas would not flow into the combustor. A simple, practical, low cost natural gas compressor is not easy to find!

Garrett's breadboard unit is operating well using low pressure natural gas as the fuel. It is meeting its performance goal of twenty-seven percent efficiency after correcting for breadboard system losses. With a larger recuperator, thirty-three percent efficiency appears feasible. Figure (3) shows a layout of the turbomachinery and Figure (4) shows the actual components.

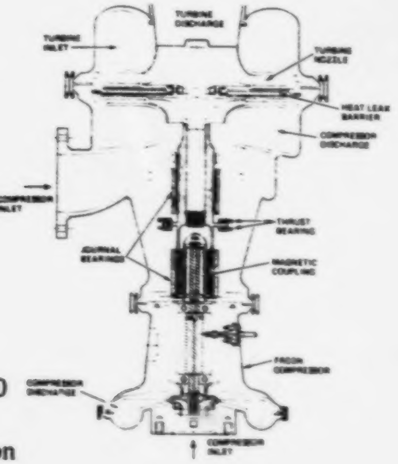


Figure 3 - AGA 10 Ton Rotating Group Cross Section

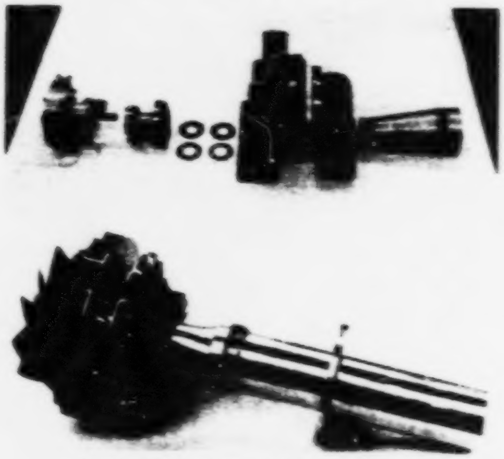


Figure 4 - Actual Components

By gas turbine standards the components of this twelve horsepower engine are very large. The turbine wheel and compressor are approximately four inches (10.2 cm) in diameter. The reason is that the working fluid density is low. The advantage in a small engine is that it is easier to make large efficient components than small efficient components. Compressor and turbine efficiencies in this engine are projected at 79% and 86% respectively. Figure (5) shows the overall efficiency and how it can be increased by improving the regenerator effectiveness or raising the turbine inlet temperature.

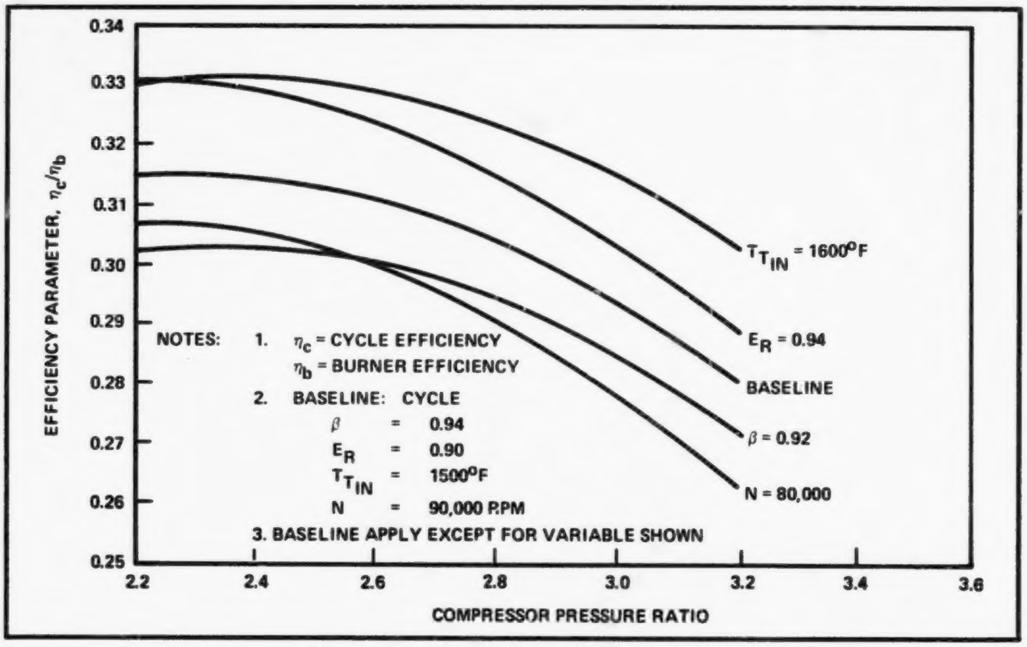


Figure 5 - Brayton Cycle Parametric Study

Fluid process bearings are used on this machine. They operate on the air used as the working fluid. They are self-pumping and require no compressed air. Most important, they require no lubrication, so this is one of a very few engines with no oil cooler, no oil pump, no oil filter and, indeed, no lubrication system. These bearings are similar to those used in the DC-10 air conditioning system where millions of hours of successful operation have been accumulated. Figure (6) is a photograph of the heat pump engine bearings.

The problem of generating power from waste heat is not significantly different from that of generating power when gas is burned at atmospheric pressure. As part of the glass furnace program discussed below, hot gas will be taken from a glass furnace and passed directly into a subatmospheric gas turbine with characteristics similar to that of the heat pump prime mover. There will be no combustor. The objective here is not to prove that the engine will work thermodynamically in this mode but to determine the practical operating problems such as the effect of glass furnace stack contaminants on turbomachinery.

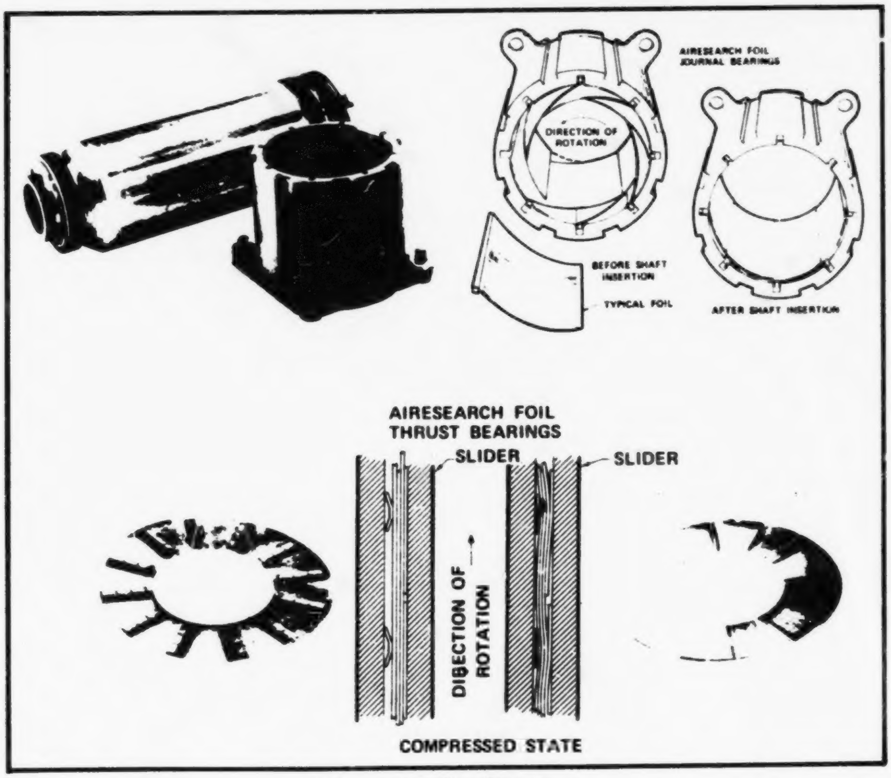

AIRESEARCH FOIL
JOURNAL BEARINGS
DIRECTION OF ROTATION
BEFORE SHAFT INSERTION
TYPICAL FOIL
AFTER SHAFT INSERTION
AIRESEARCH FOIL
THRUST BEARINGS
SLIDER
SLIDER
DIRECTION OF ROTATION
COMPRESSED STATE

Figure 6 – Gas Bearings

GLASS FURNACES

The glass furnace will be one of the first applications of a subatmospheric gas turbine to generate electricity using waste heat at 1650°F (899°C) as the energy source. The machine is nominally rated at 300 KW. The heat rejected from the engine is at 1000°F (538°C) and will be used to preheat the furnace combustion air.

The major components for this engine are being developed by Garrett under subcontract from the Alpha Glass Company who are developing the complete system under contract from the Department of Energy.

Materials used to make glass melt at temperatures in excess of 2800°F(1538°C). The products of combustion therefore leave the glass tank at temperatures in the same range. To conserve thermal energy most furnaces in the glass container industry are equipped with stacks of bricks called checkers on both the combustion air inlet and the exhaust sides of the tank. Typically every twenty minutes the flow of air is reversed so that the inlet becomes the exhaust and vice versa. Thus the exhaust is heating one stack of bricks while the other stack is heating the inlet air. Essentially the checkers are a periodic flow regenerator.

This is an excellent system which reduces the fuel consumption of the furnace by more than 50%. Although there are some radiation and convection losses, most of the Btu's absorbed in the checkers do wind up heating the incoming air and therefore reduce fuel consumption on a one for one basis.

Stack temperatures out of the checkers are in the 1200°F (649°C) range. With the addition of a subatmospheric gas turbine, however, hot air from the turbine system is used to preheat the checker inlet air. This reduces combustor fuel consumption and simultaneously raises the exhaust checker outlet temperature up into the 1650°F (899°C) range.

The reason for the increase in the exhaust checker outlet temperature is not immediately obvious. As noted above, the checkers are simply a periodic flow heat exchanger. If the inlet temperature on the cold side of a heat exchanger goes up, the outlet temperature on the hot side goes up, too.

To make power from the exhaust of the furnace, a subatmospheric gas turbine will be installed. As shown in Figure (7), 1650°F, (899°C) air is expanded through a turbine wheel into a partial vacuum of 3.7 psia (26 kPa) thus producing power. The air is cooled first in a regenerator and then in a heat sink to a temperature of about 30°F (17°C) above ambient. The air is now compressed back to atmospheric pressure and discharged. The power produced by the turbine is used to drive the compressor and the load. Simultaneously, the regenerator has heated atmospheric pressure air to 1000°F (538°C) and this hot air is ducted to the inlet air side of the checkers.

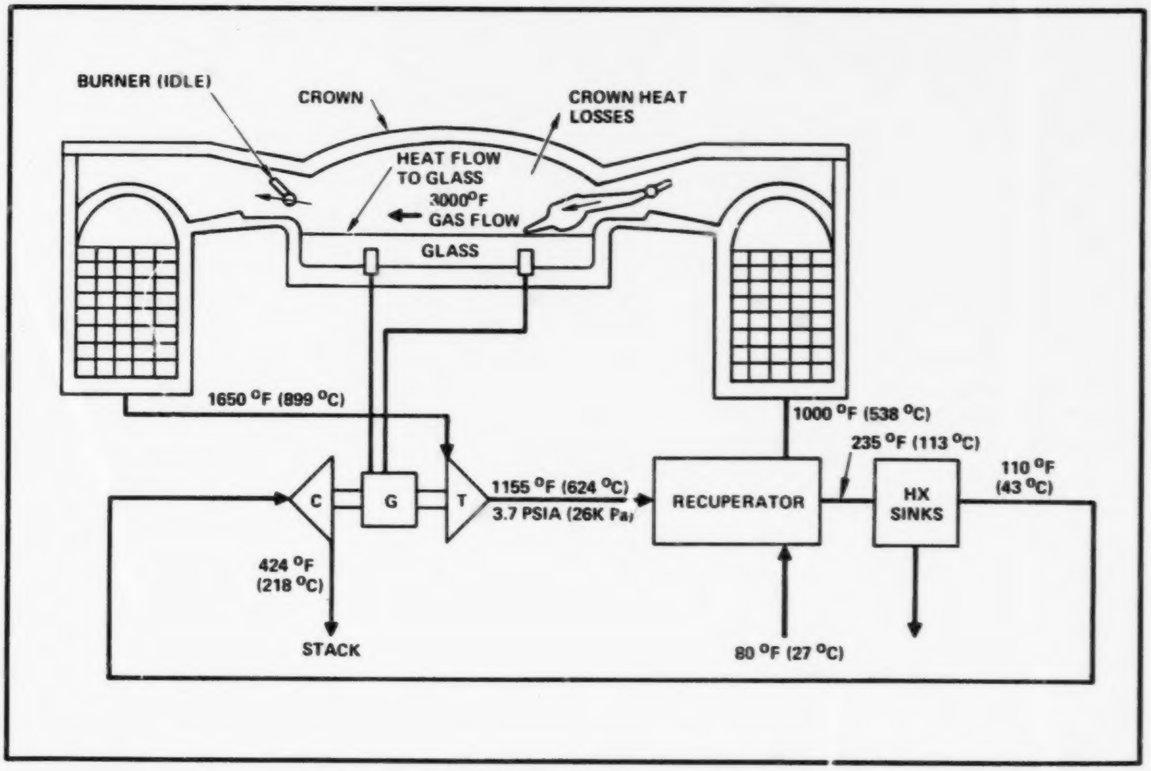

BURNER (IDLE)
CROWN
CROWN HEAT LOSSES
HEAT FLOW TO GLASS
3000°F
GAS FLOW
GLASS
1650 °F (899 °C)
1000 °F (538 °C)
235 °F (113 °C)
110 °F (43 °C)
C
G
T
1155 °F (624 °C)
3.7 PSIA (26K Pa)
RECUPERATOR
HX SINKS
424 °F (218 °C)
STACK
80 °F (27 °C)

Figure 7 – Power Genration from Glass Furnace

There are approximately four hundred glass container furnaces in the United States. A typical furnace produces two hundred tons of glass per day. This size furnace can drive two 300 KW subatmospheric gas turbine generator sets. The units require no energy other than that contained in the heat of the exhaust gases. However, as the rejected heat from the turbine is used to preheat the inlet air, overall furnace fuel consumption is reduced by more than ten percent. Thus, electric power is produced while input energy is reduced. The best of all worlds!

Figure (8) shows that as the intake checker inlet temperature goes up, the exhaust checker outlet temperature goes up and, most important, the glass production goes up, too. The increase in outlet checker temperature increases the power available which has a major economic impact. The increase in glass production due to higher furnace temperature permitting more raw material input has an even stronger economic impact. Where gas supply cannot be increased due to either gas company or government regulation, glass production can be increased by twenty percent without increasing fuel consumption.

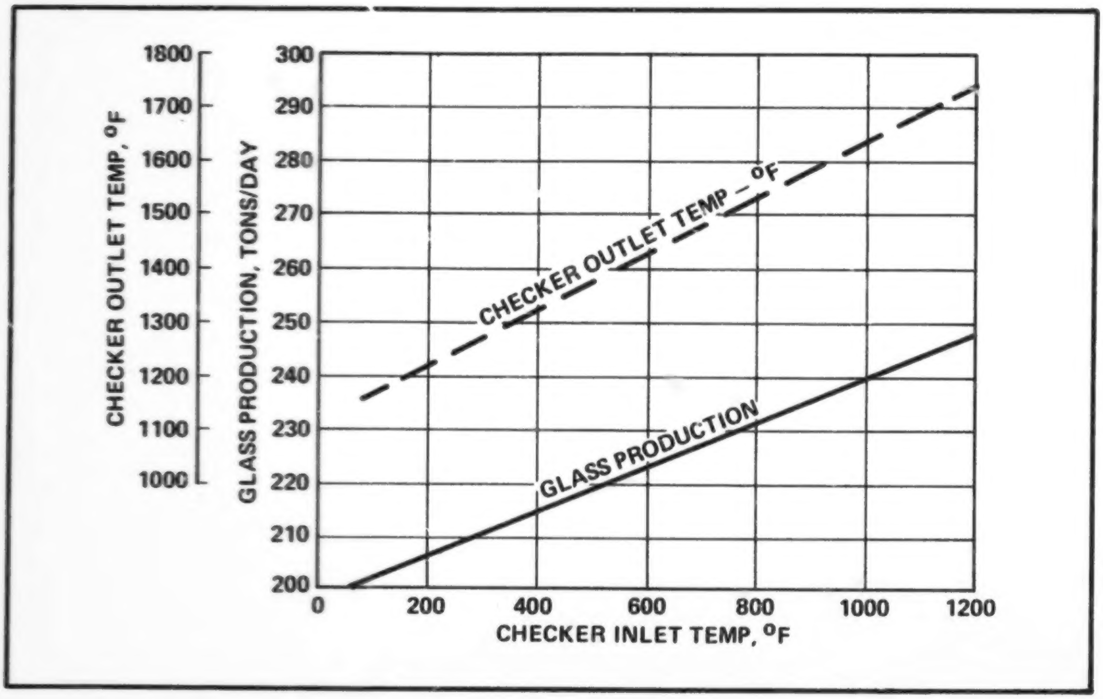


Figure 8 – Glass Furnace Characteristics

Production can also be increased by using the electrical output to power resistance heating in the tank. With molybdenum electrodes, the molten glass itself becomes the conductor and heater.

The maximum air flow and therefore the maximum fuel flow through the checkers and furnace are essentially fixed. In the first application the inlet air flow will be 584 lb/min (265 kg/min). Exhaust gas flow will be larger due to the addition of fuel in the furnace combustor and is typically 640 lb/min (291 kg/min).

Figure (9) shows the effect of pressure ratio on the regenerator inlet temperature and on the checker inlet temperature.

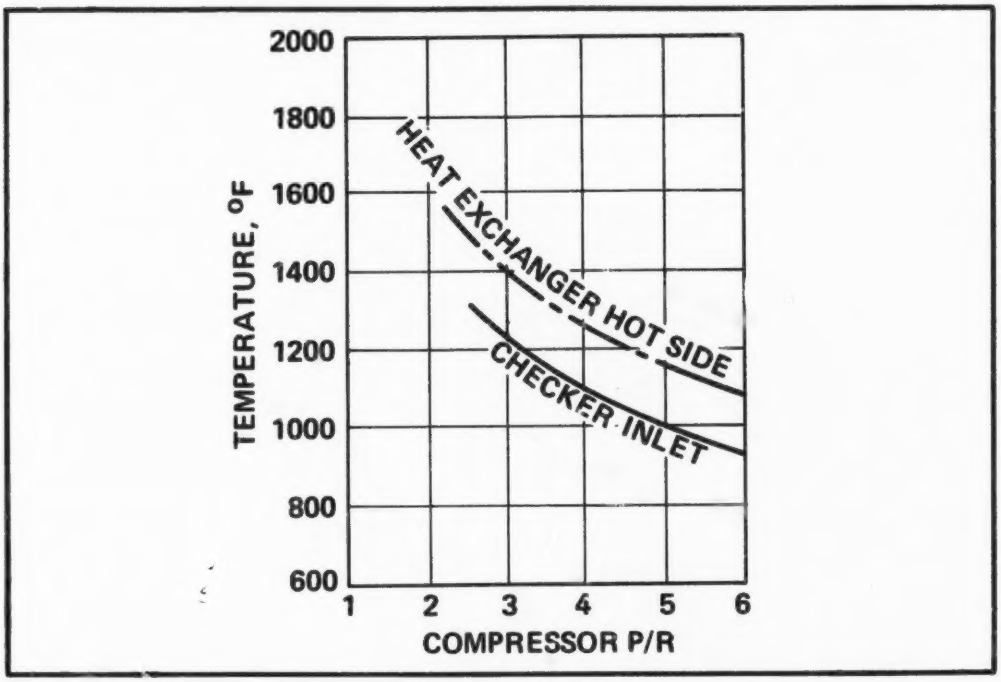


Figure 9 - Heat Exchangers - Checker Temperature Considerations.

Figure (10) shows the turbine that is being modified for this program. The turbine wheel and compressor are both radial and mounted back to back with the shaft located vertically. Figure (11) is a photo of a sister machine that could be modified to produce about 100 KW in the subatmospheric mode.

One of the keys to efficiency in a subatmospheric gas turbine is to reduce the work of the compressor. For any given compressor wheel then, the objective must be to reduce the compressor inlet temperature as much as possible, bearing in mind the cost of the heat exchanger. The present plan is to use a cross flow heat exchanger with two separate parallel passes on the atmospheric pressure side. The first pass will take ambient air in and use it to cool the turbine discharge air to about 235°F (113°C). In so doing, the ambient air will be heated to 1000°F (538°C) and this will go to the furnace inlet checker as preheated air. This portion of the heat exchanger operates essentially on a one-to-one mass ratio.

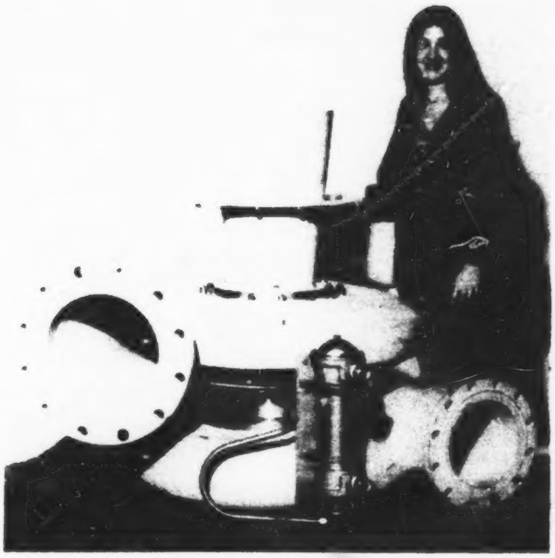


Figure 10 - 12 lb/sec Turbocompressor

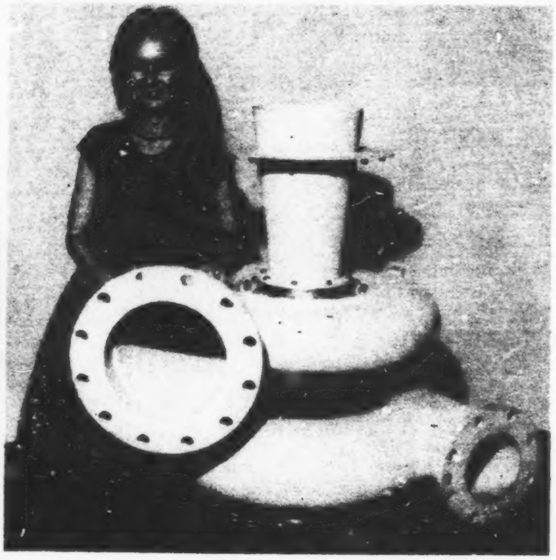


Figure 11 - 4 lb/sec Turbocompressor

The second pass will use ambient air to cool the compressor inlet air down to within 30°F (17°C) of ambient temperature. This atmospheric pressure cooling air will then be discharged into the stack.

The subatmospheric gas turbine offers a tremendous advantage in its ability to use many energy sources at atmospheric pressure and at high efficiencies. The penalty is size and weight. The heat exchanger core for the 300 KW machine would weigh 4250 pounds (1928 kg) with standard compact fins. Glass furnace contaminants will force less dense finning with a corresponding increase in weight. (These same contaminants in a conventional positive pressure gas turbine would force the heat exchangers to be tubular which would be even heavier!) The compressor wheel diameter is 18.45 inches (46.9 cm). The turbine wheel diameter is 18.75 inches (47.6 cm).

Assuming a stationary installation, size and weight are less important than cost. Cross flow heat exchangers are less expensive than counter flow. Single stage components are less expensive than multiple stage. It is expected that these units will sell for $500/KW installed which should be competitive considering the savings in energy. In a typical glass plant, these units should pay for themselves in less than two years.

We are extremely confident from a thermodynamic standpoint that this unit will work. The one major concern is that the normal carry-over from the glass tanks is abrasive at some temperatures and sticky and clogging at others. This may force the use of a tubular heat exchanger prior to the cross flow for glass furnace applications. Solutions to these potential problems are being discussed in a paper being given by Alpha Glass.

SOLAR ENERGY

The most tantalizing source of energy is solar. Volumes have been written on how it can solve the energy crisis. Yet solar energy is used effectively by nature but wasted by man.

Gas turbines are the most efficient and therefore the lowest cost near term technology for converting solar energy to electricity. Efficiency begets low cost because the majority of the cost is in the solar collectors and the quantity of collectors required for a given load is inversely proportional to the system's power conversion efficiency.

With efficient components, medium-high turbine inlet temperatures and adequate heat transfer surface, the gas turbine efficiency can be forty percent or more. The problem is to get the solar energy into the gas turbine cycle. Once again, the cost and availability of high temperature heat exchangers imposes limits. Once again the subatmospheric gas turbine gets around these limits by permitting the heated air to enter the turbine wheel directly at atmospheric pressure with no heat exchanger.

Figure (12) shows a subatmospheric gas turbine solar absorber. Note that the solar absorber is completely unpressurized. There are no seals and no mechanical constraints to thermal growth. The absorber surface may well be sectioned to prevent uneven heating from causing uneven growth.

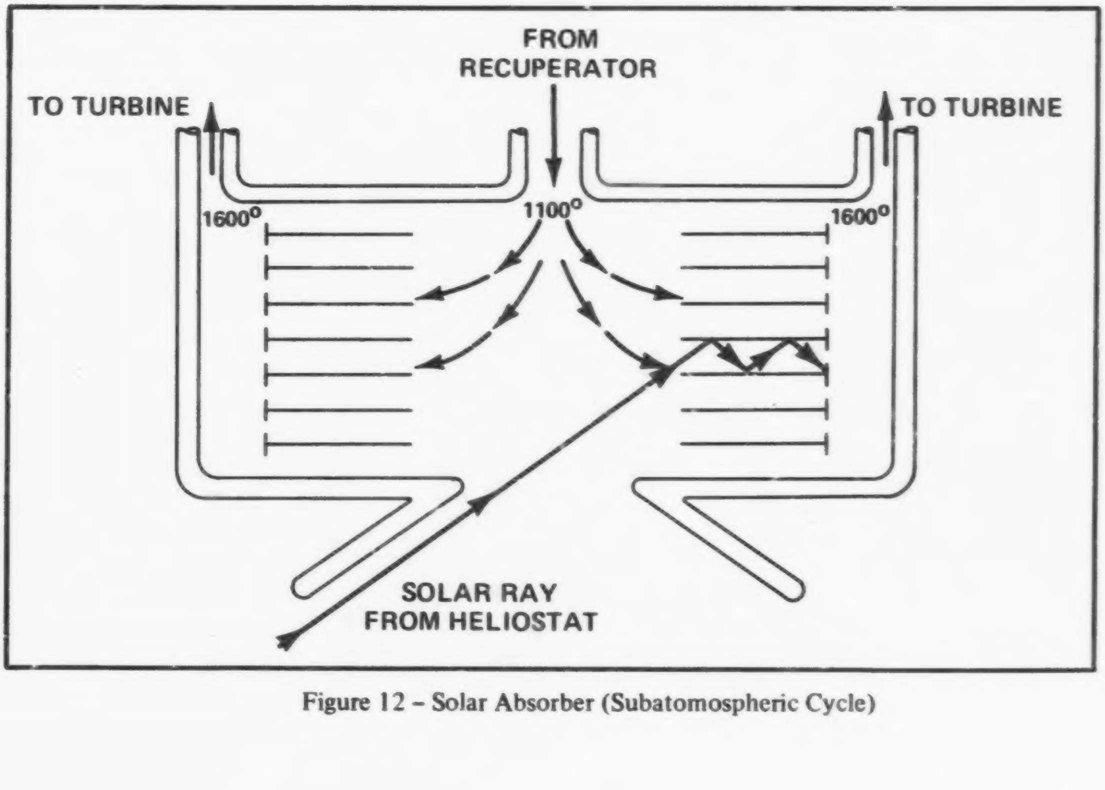
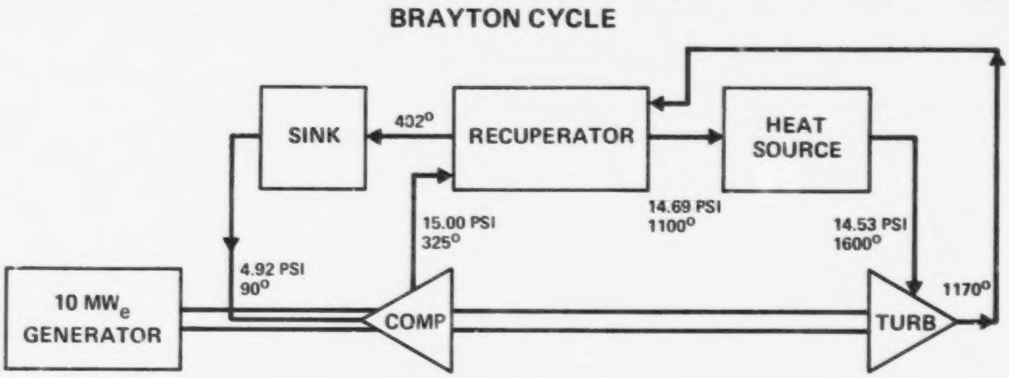


Figure 12 – Solar Absorber (Subatomospheric Cycle)

The same surface absorbs the sun's radiation and heats the turbine's cycle air by convection. Therefore the thickness and thermal transmissivity of the material are of no consequence. Heat is not transmitted through the material.

In the concept shown it is assumed that the surfaces first struck by the sun will be slightly reflective. The objective is to have the hottest part of the absorber at the point where the heated gases leave. This maximizes the absorber's efficiency and decreases the reradiation.

Preliminary sizing has been done on a ten megawatt turbine. As shown in Figure (13) the components are large. Indeed, they are so large that they rotate at only 3600 rpm, thus eliminating the need for a gearbox. The key element, however, is the efficiency. This machine should operate at over 43% efficiency and can be pushed to over 46% with extended heat transfer and reduced internal pressure losses. With organic Rankine bottoming, over 50% efficiency should be possible. This high efficiency would reduce the quantity of collectors required substantially.



WEIGHTS	
RECUPERATOR	532,000 LBS
SINK	5,000 LBS
WRAP-UP	40,000 LBS
TOTAL (HT EX)	577,000 LBS

MISCELLANEOUS	
TURBINE DISCH DIA	104.1"
TURBINE DUCT DIA	134.8"
MASS FLOW	163.0 LBS/SEC
SINK FLOW	375.0 LBS/SEC
SINK FAN POWER	170 HP

EFFICIENCIES	
GENERATOR	0.98
RECUPERATOR	0.925
COMPRESSOR	0.872
TURBINE	0.914
SINK FAN	0.7
CYCLE	0.432

Figure 13 – Subatmospheric Gas Turbine

The subatmospheric engine is not limited to power towers. A derivative of the ten-ton heat pump engine would be an ideal candidate. Figure (14) shows one module of a dispersed system using a 23 ft (7m) dish with an absorber and an 8 KW subatmospheric gas turbine at the focus.

The quantity of modules can be varied to match any size load. The modules themselves can be installed in a wide range of locations such as rooftops or under power transmission lines where central power towers are impractical.

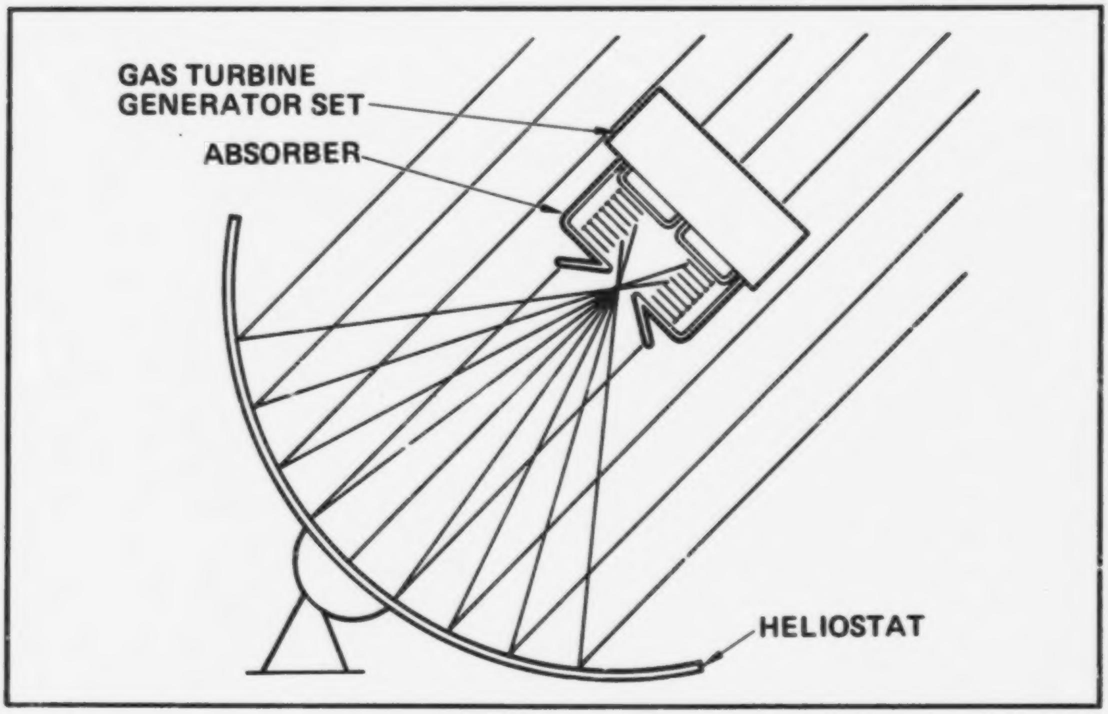
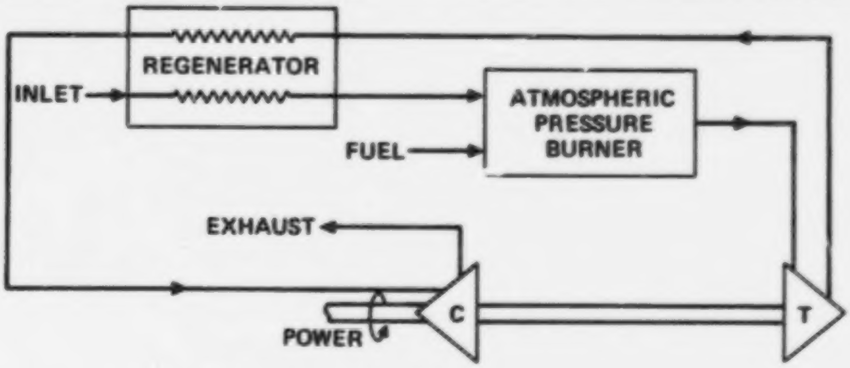


Figure 14 - Complete Module

LOW BTU GAS

Low Btu gas is available from sewage disposal plants, solid waste disposal, mines, oil fields, etc. The difficulty in burning low Btu gas in a gas turbine is that it must be compressed to a pressure higher than that in the combustor or it will not flow in. The lower the Btu content, the more gas necessary, and the more horsepower lost in fuel gas compression.

In the subatmospheric gas turbine, no fuel gas compressor is needed. The fuel is burned at atmospheric pressure, as shown in Figure (15).



- **LOW BTU GAS – MINES**
  - **– OIL FIELDS**
  - **– SEWAGE DISPOSAL**
  - **– SOLID WASTE DISPOSAL**
- **BIOMASS – WOOD CHIPS**
  - **– AGRICULTURAL WASTE**

Figure 15 – Other Uses'

BIOMASS

Wood chips, agricultural wastes and other biomass have enormous potential for electricity generation. As long as they are burned to produce temperatures in the 1500°F (816°C) range or higher, the hot gas produced can be used directly without compression in a subatmospheric gas turbine to produce power.

POLLUTION

One of the key considerations of any prime mover with a combustor is the NOX output. Although detail tests have not been conducted, there is every indication that NOX formation will be substantially lower in the subatmospheric gas turbine than in the conventional gas turbine because combustion takes place at atmospheric pressure.

SUBATMOSPHERIC GAS TURBINE VERSUS STEAM TURBINES

Both the subatmospheric gas turbine and the steam turbine can be powered by the energy in hot gases at atmospheric pressure. It matters little whether these hot gases come from processes, engines, combustion of fuels or solar energy. The subatmospheric gas turbine, however, offers certain advantages over the steam turbine.

1. Because higher cycle temperatures can be used when the source temperatures are high, the Carnot efficiency is better, resulting in higher actual efficiencies.
2. Because efficient gas turbine components are available in low power sizes, the cycle efficiencies in small engines are much better.
3. Because the heat rejected is in the 1000°F (538°C) range as opposed to the very low heat rejection temperatures of the steam turbine (a) better use can be made of the rejected heat in a process, (b) bottom cycles are feasible and (c) regeneration is feasible.
4. Because the heat rejected is in the 1000°F (538°C) range in a simple cycle or 300°F (149°C) range in a regenerated subatmospheric cycle, air cooled heat sinks are used. With a counter flow heat exchanger, the heat sink and cooling air flow can be quite small. With Rankine cycle, the condensing temperature determines the maximum temperature of the cooling air flow. The low temperature difference demands a large condenser and a large air flow. Otherwise, water cooling must be used which poses other problems in many areas.
5. Because no part of the engine operates at a pressure in excess of a few inches of water pressure, most codes will not require an operator.

CONCLUSION

We are extremely optimistic on the future of subatmospheric gas turbines. They are highly efficient prime movers capable of receiving energy from a wide range of sources. The heat pump and the glass furnace programs will be the first practical applications. These programs offer promise of successful production machines that should see uses far beyond those envisioned today.

FOR ADDITIONAL INFORMATION

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GARRETT

WASTE HEAT RECOVERY FROM INDUSTRIAL PROCESSES USING BRAYTON CYCLE TURBINE TECHNOLOGY

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ABSTRACT

This paper describes a method of increasing the efficiency of a variety of energy intensive industrial furnaces by up to 50% through the application of hot gas turbine technology being developed by Alpha Glass, Inc. and Garrett AiResearch.

A conservation demonstration program is underway to validate the concept of using Brayton cycle turbine technology for energy recuperation, generation and displacement. Initial system design uses validated materials and components subsequent to technology validation. The system is intended to be complementary to other energy conservation methods. Though applicable to many industrial environments, the glass industry is specified as the prime candidate for use of this technology and for the supporting economic and technical risk analyses. A cooperative demonstration program is being conducted by Alpha Glass, Inc. with a Norton Simon Industries' subsidiary, Glass Container Corporation, at their Vernon, California plant (Figure 1).

INTRODUCTION

Industry accounts for about 40% of total energy use in the United States. Of this amount, one-third to one-half of the heat energy developed in fuel combustion processes is lost in the form of waste heat through radiation or exhausted gases.

There is a very real opportunity to increase the efficiency of fuel use in many industrial processes through recovery and re-use of this waste heat, either for application in its raw form or conversion to other forms of energy - such as electricity.

The glass industry, in particular, uses large amounts of heat energy in the melting and forming of their raw materials into end products. This industry was selected to demonstrate the possible efficiencies attainable through

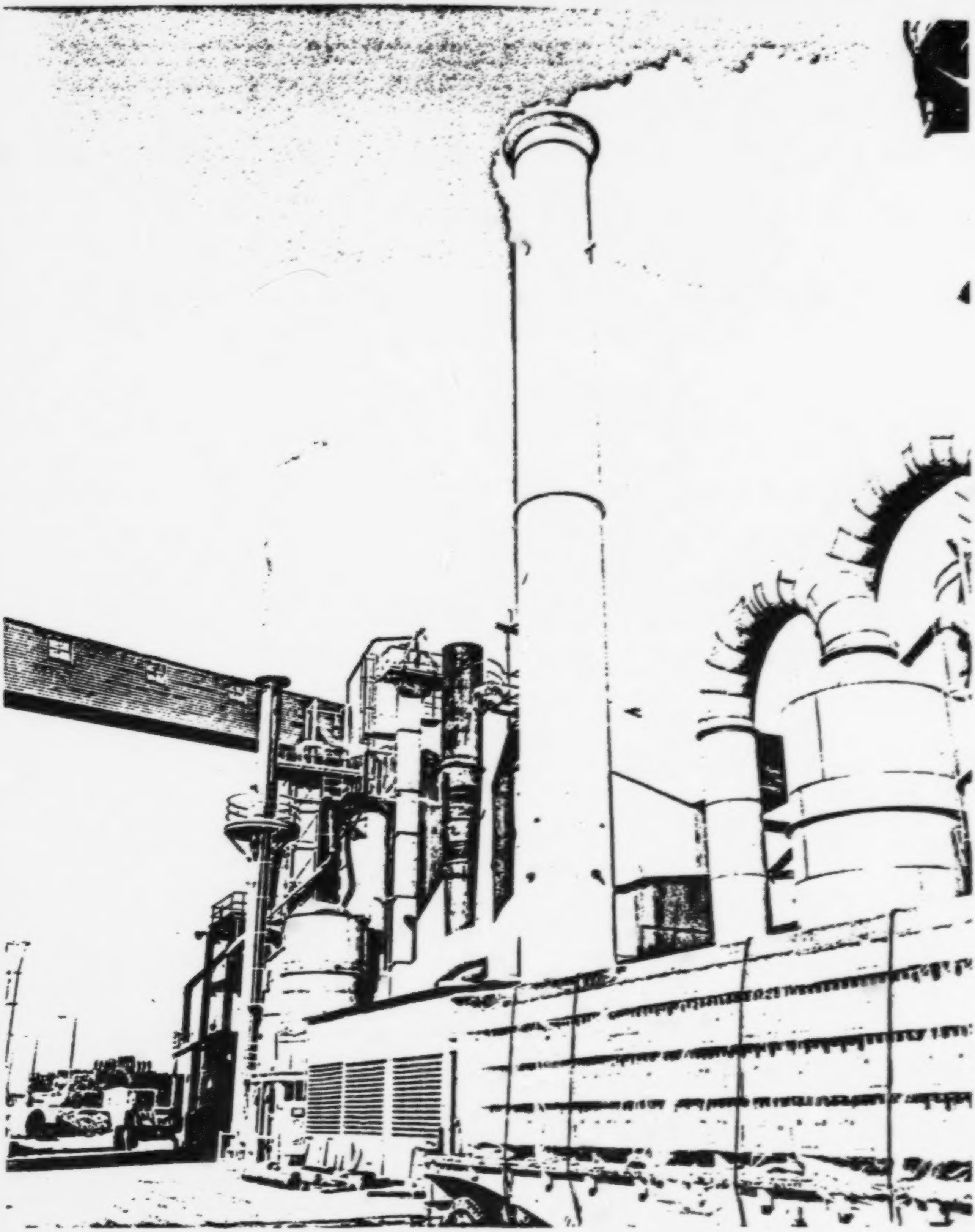

Figure 1. Glass Containers' Plant at Vernon, California

application of a subatmospheric Brayton cycle turbine to recover and convert high temperature gases that would otherwise be lost up the exhaust stack. The particular concept, engineering and installation are attributable to Alpha Glass, Inc. Energy Research & Applications, Inc. (an Alpha Glass, Inc. affiliate corporation) has been contracted to perform economic and efficiency analyses, environmental impact assessment, and determination of applications to other industries.

Simplistically, the key feature that distinuishes a Brayton cycle turbine is operation at constant pressure without an intermediate working fluid, i.e., waste heat in stack gas is used directly. Temperature in, less temperature out generally determines the efficiency of any heat engine cycle. Theoretical efficiency (discounting mechanical efficiencies, etc.) is very favorable in the Brayton cycle compared to the Rankine cycle at the high temperatures ($\geq$1000°F) of a waste gas stack. Since the major goal of this activity is to demonstrate efficiency improvements, the Brayton cycle technology appears to be a reasonable choice for industrial users with high temperature waste gas streams.

Glass Industry Operations

The glass industry is a very old one whose product output has expanded to major proportions in recent history. Glass containers (bottles, jars, etc.) such as being produced at Glass Containers Vernon facility (figure 2) account for over 15 million tons of production each year, while flat glass and blown glass (such as fiberglass) account for another three million tons each. Additionally, specialty glasses - such as those for optics - are continually produced though at lower yields and in different types of glass melting tanks.

Although glass is a very versatile product, with uses ranging from soda pop bottles to electronics, all glass forming furnaces perform basically the same functions. This makes the application of energy recovery/conversion technology even more attractive since the components can be, to a large extent, standardized. Because of current and projected increases in energy costs, all glass-making operations can benefit from innovative conservation technology.

Glass furnaces have already experienced many efficiency improvements in arriving at their present operating configurations. Fine tuning and optimum location of burners, port location and better refractory shapes are examples of refinements already implemented.

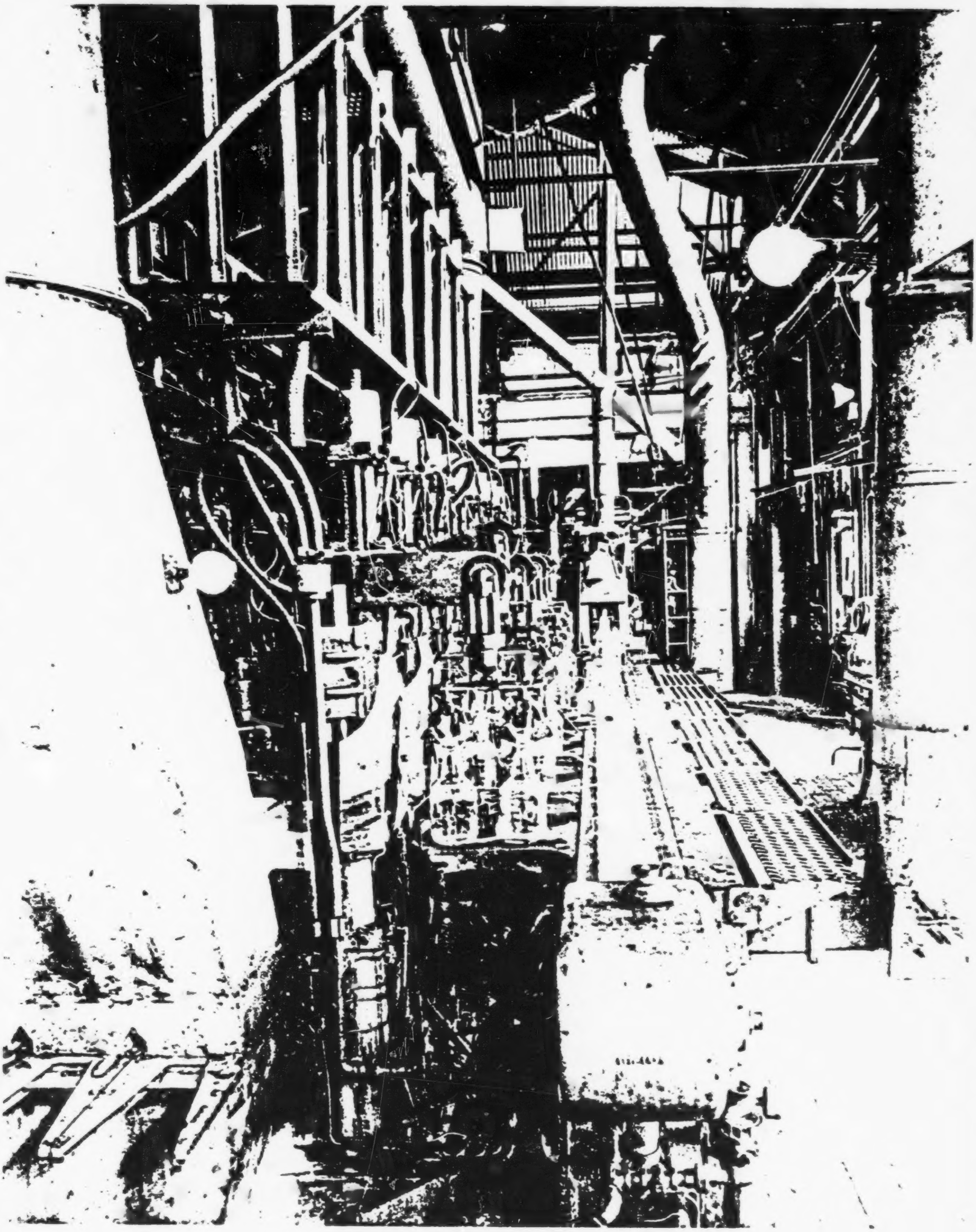

Figure 2. Bottle forming Line

Today, most furnaces also have ceramic heat regenerators known as "checkers." The checkers absorb waste heat while the glass furnace fires in a given direction; then returns some of the waste heat to the incoming combustion air stream when the furnace reverses firing direction. Prevailing furnace reversal cycles average 20 minutes. During the reversals, waste heat temperatures decrease for up to two minutes.

The regenerative furnaces are used primarily in the container and flat glass segments of the industry. These categories of glass product represent over 80% of total industry tonnage. As the glass industry's fuel has historically been natural gas, any efficiency improvements look especially attractive.

Applications of the Subatmospheric Brayton Cycle Turbine

Subatmospheric Brayton Cycle technology has been determined applicable to two of the furnaces utilized in preparing raw materials for glass production and to all of the furnaces used in the container, pressed, and sheet glass processes.

A range of cost savings for various glass production activities has been estimated as follows:

Glass melting:	21% to 42%
Glass forming and production:	10% to 30%
Synthetic soda ash production:	20% to 30%

The savings are due entirely to the consequent reduced fuel costs realized through improved furnace operating efficiencies.

Application of the technology to other energy intensive industries has been determined feasible and some preliminary estimates of fuel costs savings made. Hardware component installation is, of course, different for the various industries considered, but the same system components are involved. The glass industry is actually a good example of a worst case test of system implementation due to the very restricted volumetric space for equipment and high working environment temperatures in most glass plants. Percentage savings in some other industries realizable through Brayton cycle turbine technology are eastimated as:

Coke production:	20% to 30%
Steel:	10% to 15%
Zinc smelting, refining:	20% to 30%
Copper smelting:	40% to 50%

While the furnace interface requirements are industry-sensitive, the general technology is applicable to any furnace with exhaust gas temperatures of 1100°F or more. The industries listed above consume 403 x 10^{12} BTU of purchased fuels and electricity each year, based on available figures. Considering all industries where the Brayton cycle turbine technology could be used to increase efficiencies and assuming a minimum of 10% application, potential energy savings amount to 216 x 10^{12} BTU/year on a national level.

Examples of types of furnaces (other than glass melting) that can realize increased fuel efficiencies through Brayton cycle turbine technology are open-hearth furnaces, distilling furnaces, rotary kilns, and basic Bessemer converters. Detailed economic and environmental analyses are being performed using the glass industry case as an example. There is good transference of derived data in some areas: most obviously, cost per ton of product has the additional benefit of increasing price competitiveness of the final product in the marketplace.

Conservation Options in the Glass Industry

Preliminary analysis of conservation options related to heat transfer reveals two general categories of activity:

1) Reduction of furnace heat loss, and
2) Improvement of heat transfer to the product by the furnace.

Category 1 normally is independent of Category 2. All cost-effective possibilities in that category should be exhausted to make a furnace as efficient as possible as a baseline operating condition prior to examining options in Category 2. Static efficiency improvements can be realized by insulating and reducing air infiltration to the furnace.

Four alternatives in Category 2 were selected for further examination of effects and tradeoffs in implementation. They are:

1) batch preheat using direct waste heat,
2) create H_2O steam with waste heat for power generation,
3) create organic steam with waste heat for power generation, and
4) generate power directly from waste heat using a subatmospheric Brayton cycle turbine.

The last option proved to be the most promising technology for demonstration and provided the best economic return to the glass industry.

Brayton Cycle Turbine System Configurations

There are several possible physical designs and interfaces possible with Brayton cycle technology. This flexibility in itself makes the method useful for the broadest range of applications.

Four system designs were initially defined for use with the glass industry's furnaces. Engineering specification of each design was completed and the four systems were evaluated according to criteria of efficiency improvement versus cost, economic returns and risk versus investment. Suitability for subsequent system enhancement was also considered. The four designs for Brayton cycle turbine application have been designated System A, B, C and D. The distinctive characteristics of each system are briefly summarized in the following paragraphs.

System A (Positive pressure system using stack gas)

This system is designed to interface in the waste heat stack of the glass furnace. A heat exchanger in the stack is used to transfer the waste energy from the melting tank to the turbine. Energy from System A is used to generate compressed air in a form deliverable for glass container forming in the plant. Residual heat is recycled into the furnace as preheated combustion air.

System B (Subatmospheric system using stack gas)

System B interfaces with the furnace at the glass tank checkers reversing valve. This results in waste heat from the tank being directly introduced into the turbine. The gas is further cooled by a heat exchanger and heat sink before being admitted to the turbine compressor. Shaft horsepower from the turbine is transformed into electricity through a generator and reintroduced into the glass tank through electric "boosting" of the melt. This technique permits substantial production increases from a given furnace. Residual heat is recycled to the furnace as preheated combustion air.

System C (Subatmospheric system using melting tank exhaust)

System C interfaces with the glass furnace at the top of the checkers or directly with a glass furnace not having checkers, such as a unit melter. This system operates as System B but requires dilution of the waste heat stream to 1650°F.

System D (Subatmospheric high temperature system using melting tank exhaust)

System D interfaces directly with the glass melting tank and operates as System C. However, the tank waste gas is not diluted prior to entering the turbine. This results in improved engine efficiency and consequent higher furnace efficiencies. Continuous operation at temperatures up to 25°F requires development of new, probably ceramic, materials for turbine blades and recuperator.

In general, Brayton cycle turbine systems present an opportunity to recover waste energy and deliver power in a variety of forms as dictated by end use requirements. This energy can displace an energy form being provided by other plant equipment, including shaft horsepower, compressed air, hydraulic pumping, cooling air and electricity.* Savings in BTU's required for an industrial process that are achievable through such an energy conservation system are often a factor of three or more above the electric energy rating of other equipment that has been conventionally used to generate power.

System Value

Of the various energy forms listed in the previous section, 60-cycle electricity is the most universally used, and has therefore been selected to evaluate the energy savings achieved through Brayton cycle waste heat recovery. A multi-variate economic model has been constructed to quantify the annual savings possible through recovery of waste heat, and to establish the maximum investment in equipment which will produce any required after-tax rate of return on the investment. The use of multiple variables makes it possible to change any of the baseline data, which are considered to be conservatively realistic. In the baseline example, a subatmospheric system produces 600 KW of electricity continuously and saves 10% of the glass tank's fuel consumption of 1.2 trillion BTU's per day (in a 200-ton per day tank). Electricity is valued at 3.4¢/KWh (a combination of energy and demand charges) and fuel at \$2.00/MBtu. Other input variables include inflation (10%), maintenance and overhaul (\$25,000/year), project life and decision time horizon (7 years) and required after-tax internal rate of return (25%). A simplified nomograph illustrates the relationship between annual cost saving and justifiable investment, with the resulting maximum investment for the baseline example equalling $739,000. Substantially affecting this investment result is the assumption that both a 10% energy tax credit and a 10% investment tax credit will be granted.

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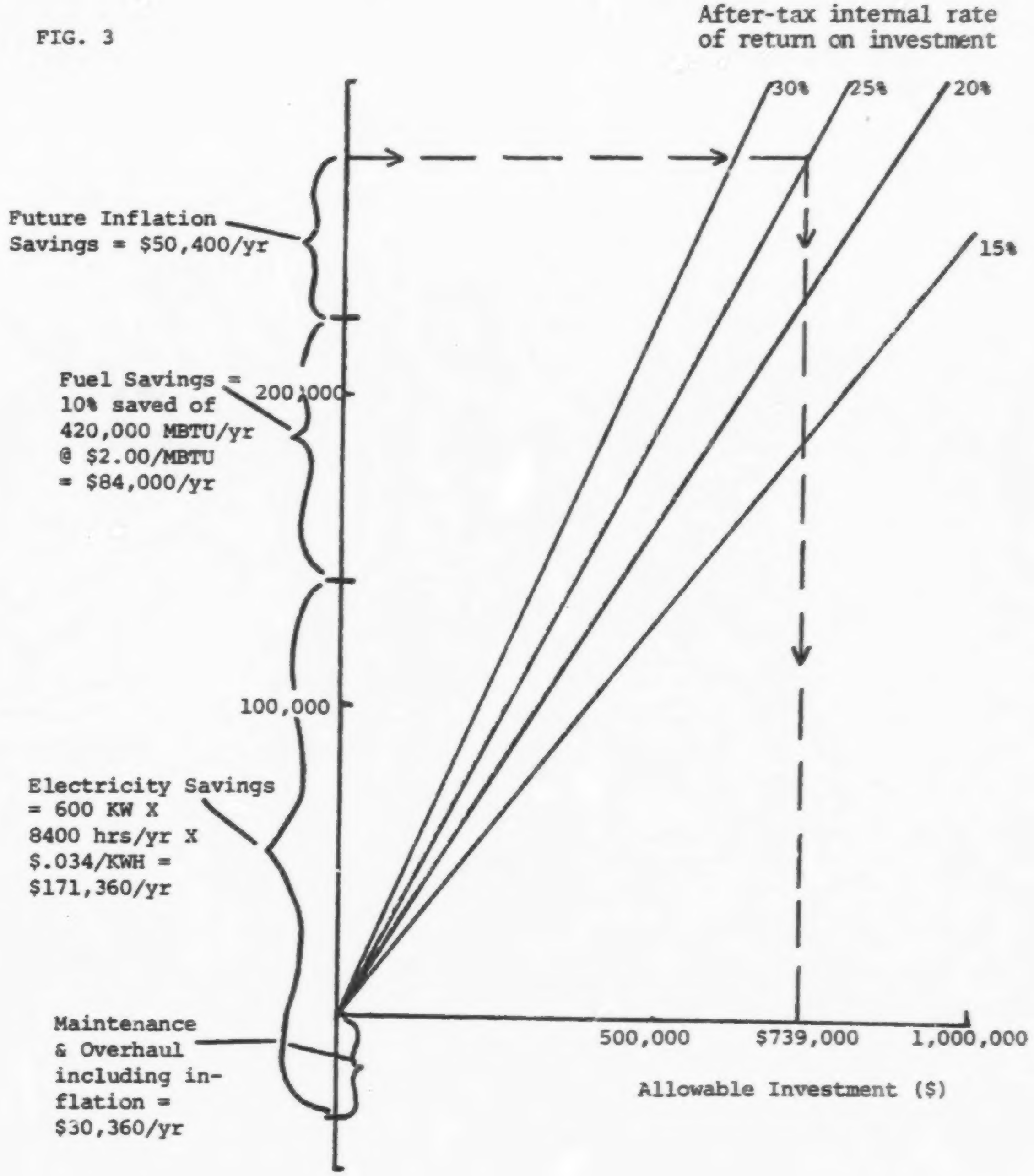
*The electricity is not necessarily 60-cycle; other frequencies can be produced.

FIG. 3



27 July 1978

ASSUMPTIONS IMPLIED BY THE MODEL

BENEFITS - Presumably the power generated by the waste heat recovery system can all be used within the plant, so no sale of electricity will arise.

- The glass plant and waste heat recovery system operate 24 hours/day. The added savings due increasing prices of fuel and electricity are calculated for nominal price changes, as are increases in maintenance and operation costs.

- All annual savings represent additions to taxable income, and hence increase income taxes.

- Unquantified benefits may include reduced susceptibility to utility power failures & possibly improved pollution abatement.

COSTS - Depreciation of the new system is straight line over seven years, applied to 100% of the gross installed cost.

- Property taxes are levied on the new equipment at a rate of 1% per year, applied to 100% of the gross installed costs.

- Aggregate income tax rate is 50% on taxable income, with deductions allowed for maintenance and operation, depreciation, and property taxes.

- Installation is assumed to be included in system purchase price with labor performed by a contractor, enabling the full installed cost of the system to be claimed for tax credits, depreciation, and property taxes.

- The firm will receive an investment tax credit of 10% and an energy tax credit of 10% (both applied to full installed system cost) at the end of the year of purchase. These credits require that the firm's tax liability exceed the amount of credits.)

- No downtime for glass production is anticipated during installation of the waste heat recovery system.

- The space occupied by the new system is assumed to have no value in alternate uses.

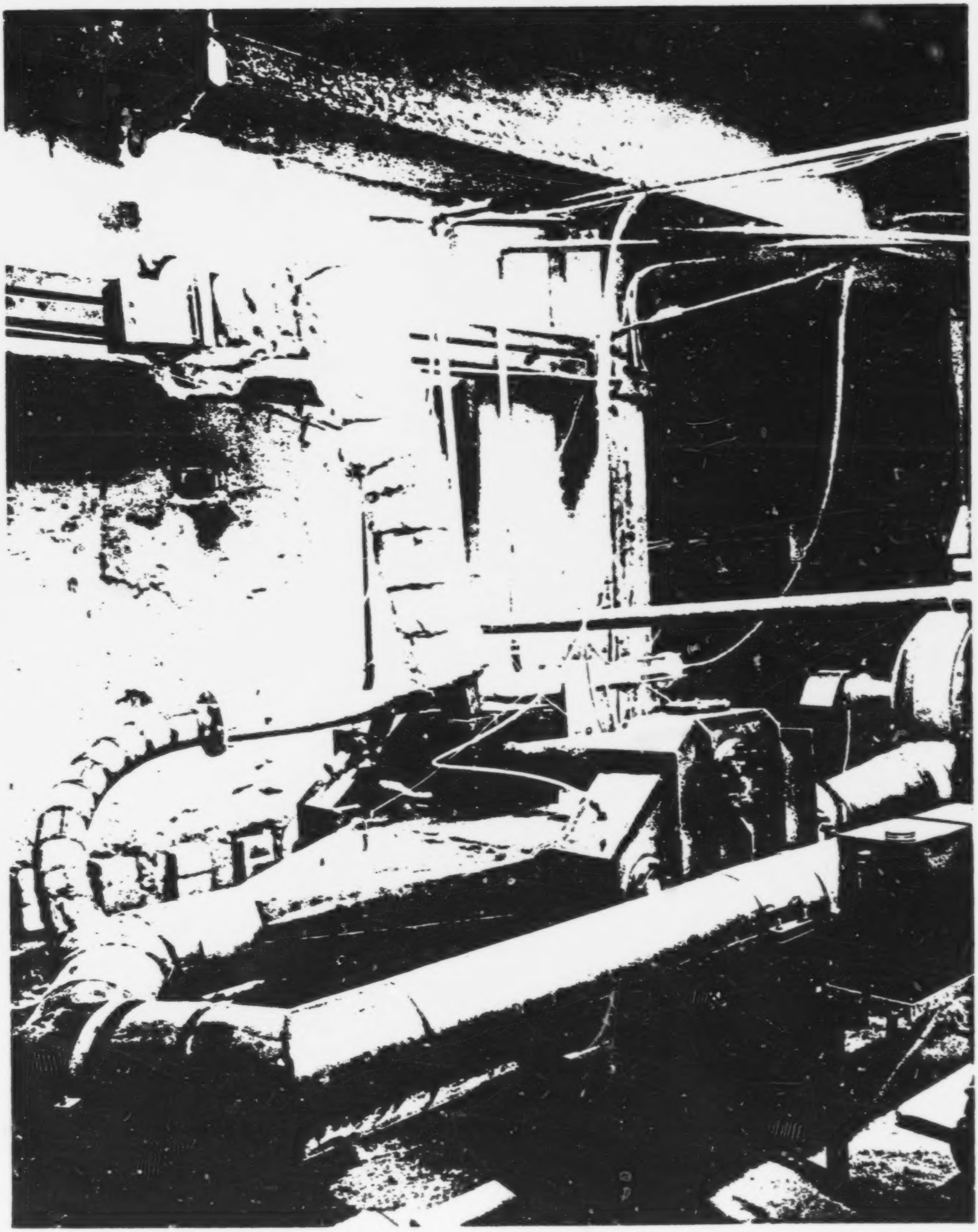


Figure 4. Subatmospheric Brayton Cycle Test Unit.

The economic model lends itself to computer programming and has been conveniently formulated in an input-output program which quickly determines the investment a firm can afford to make in a system which produces various assumed cost savings.

Furnace Modification

In preparation for the on-site testing of the Brayton cycle subatmospheric waste heat recovery system, several modifications have been made at the demonstration glass plant. Probes and sensors, with appurtenant duct work, have been installed on the checkers in order to monitor temperatures, NO_x, SO_x, and particulates. Figure 4 shows the subatmospheric system installed and operating at the demonstration site, including ducting and support structures.

System Test

The key question to be examined in the test program concerns the behavior of impurities in the waste gas stream at different temperature levels. Tests to date have indicated that the heat exchanger (a plate-fin type) will become plugged with the waste gas impurities at temperatures in the 1250°F range, as pictured in figure 5, but will not plug at lower temperatures of 900°-1000°F. The exact temperature at which plugging of the heat exchanger occurs is currently being determined experimentally. Several solutions and remedies have been considered to deal with the impurities in the gas stream, including mechanical cleaning devices (chains, screens, scrapers, etc.); chemical cleaning methods (limestone filter); reconfiguring the system in such a way that the troublesome temperature range is avoided (see the following section on system configuration); and application of special materials for the hardware.

The two most promising approaches to cleaning the heat exchanger without severely compromising the thermal efficiency of the system at present are: 1) a periodic air cannon blast on the plate-fin heat exchanger, which sends a sonic blast through the device, or 2) the use of a hybrid dual heat exchanger comprised of a plate-fin heat exchanger following an annular shell heat exchanger, the latter to be cleaned with a suitable mechanical device. The air cannon is a more attractive alternative on volumetric and economic grounds, and therefore is currently the subject of experimentation, but has not yet been sufficiently tested in the glass plant environment.

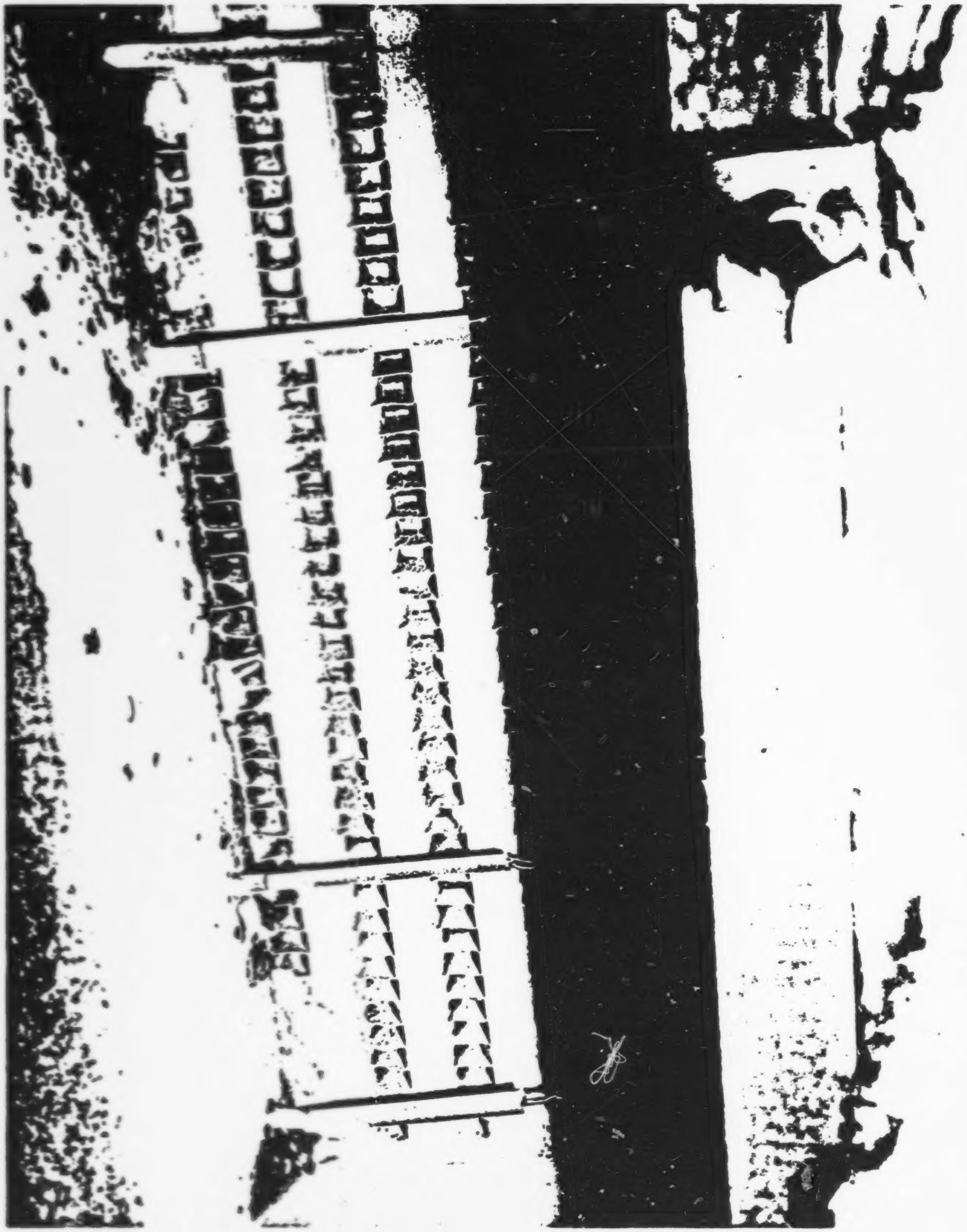


Figure 5. Plugged Heat Exchanger.

Model Furnace

A scaled-down glass furnace has been constructed at Alpha Glass in El Segundo for purposes of running experimental tests in a controlled environment. The model furnace will be used to test the turbomachinery hardware components in a realistic, controlled operation. The proposed heat exchangers will also be tested and monitored, along with the glass-making operation itself, in order to optimize the frequency and level of electric boosting in the furnace. The use of a small (one and a half tons per day) glass furnace in a well-controlled and monitored environment will accelerate the collection of data on waste heat recovery and improve its accuracy. A photograph of the model glass furnace appears in Figure 6.

Alternative System Configurations

From the four (4) basic system alternatives outlined earlier have evolved eight (8) possible system configurations, all based on the same turbomachinery (shown in Figure 7). The initial subatmospheric cycle can also be operated in a positive pressure mode, and either of these two modes of operation are applicable to several areas within the glass-making operation. Figure 8 briefly sketches the eight (8) alternative cycles, and gives preliminary cost estimates for each. Cost estimates vary largely because of the different approaches used in gas-stream cleanup. The air cannon is proposed for the baseline cycles (subatmospheric and positive pressure); the hybrid dual heat exchanger is included in the costs of two cycles; and the remaining four (4) cycles avoid the plugging problem either by using clean air or low temperatures.

Further tests and study are necessary to determine which configurations will work in the glass plant environment, and of these one will be selected as the economically optimal choice.

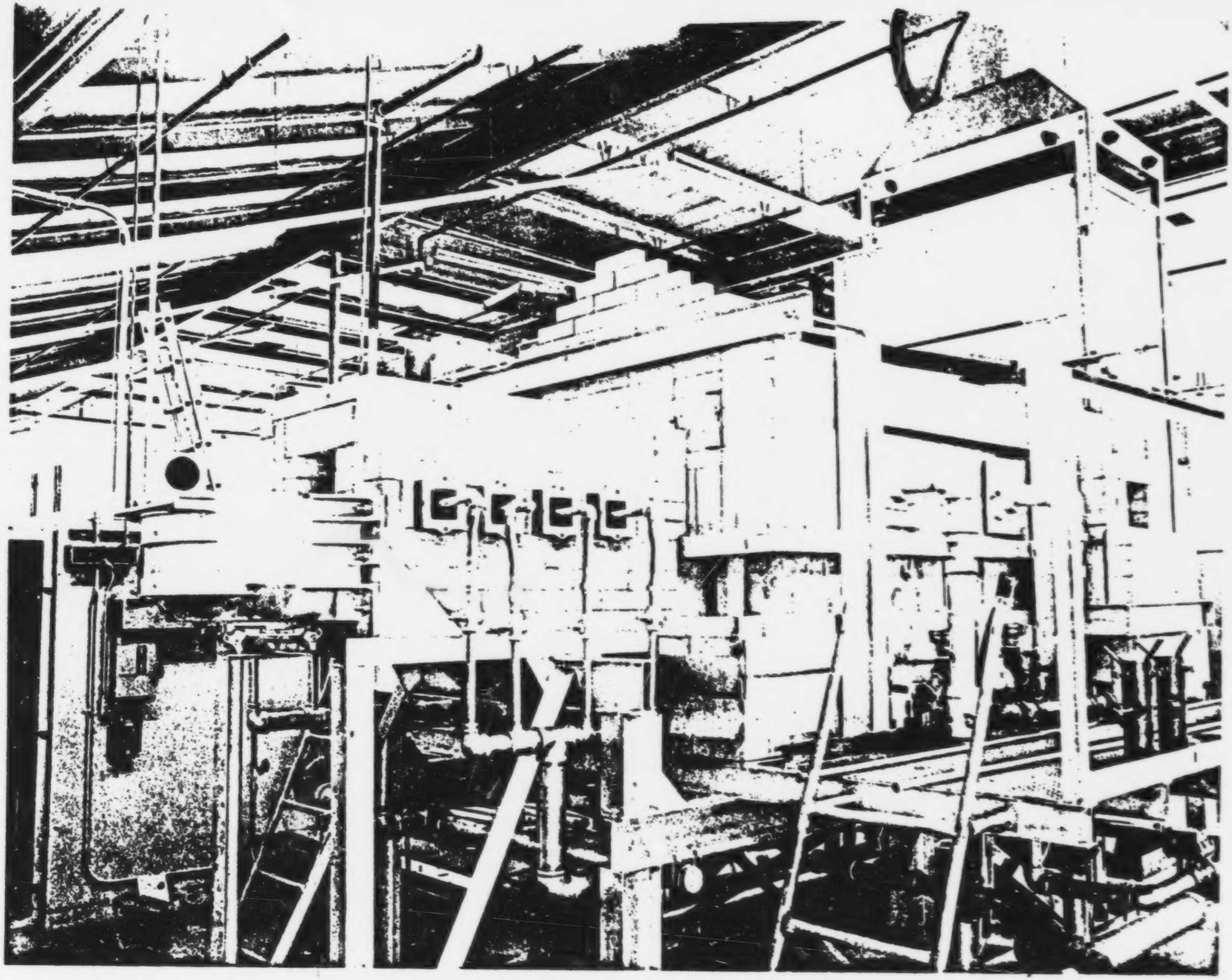


Figure 6. [redacted] Model Furnace.

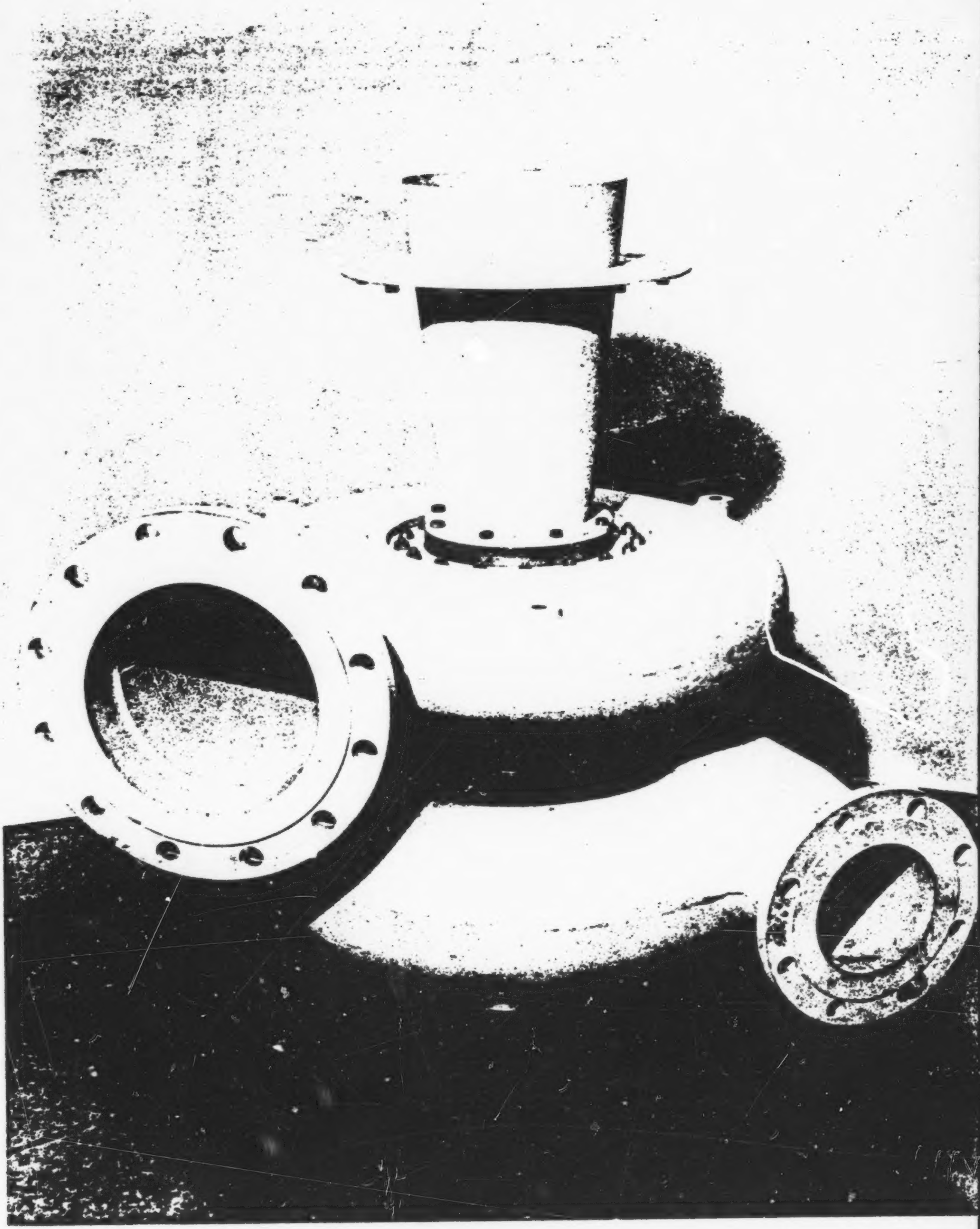
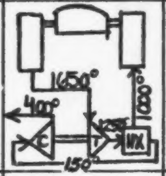
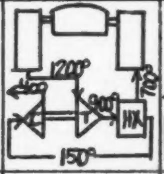
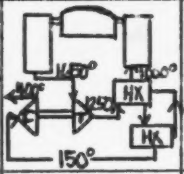
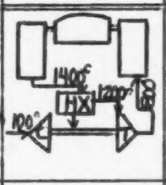
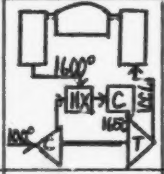
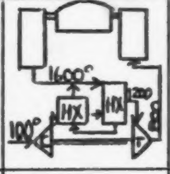
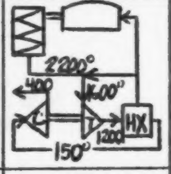
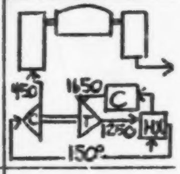


Figure 7. Representative Hardware.

	Baseline SAS	SAS Low Temp	SAS with Hybrid H.X.	Baseline PPS	PPS with Combustor	PPS With Hybrid H.X.	SAS with Clean Air	SAS Topping Cycle
FIGURE 8. PRELIMINARY SYSTEM COST ESTIMATES	 1650° 400° C T 1250° HX 1000° 150°	 1200° 400° T 900° HX 150°	 1650° 400°F 1250° HX HX 150°	 1400°F HX 1200° 100° T	 1600° HX C 1650 100° T	 1600° HX HX 1200 100° T	 2200° 400 1600° T HX 1200 150°	 450 1650 C T 1250 HX 150°
Turbomachine	80,000	80,000	40,000	40,000	40,000	40,000	160,000	80,000
Plate Fin Heat Exchanger	60,000	60,000	50,000	100,000	100,000	30,000	80,000	40,000
Radiant Tube Heat Exch.	--	--	73,500	--	--	220,500	---	---
Combustor	--	--	--	--	10,000	--	--	10,000
Controls	20,000	20,000	20,000	15,000	15,000	15,000	25,000	20,000
Switch Gear	8,000	8,000	8,000	5,000	8,000	5,000	14,000	8,000
Wrap-Up	60,000	60,000	75,000	100,000	100,000	150,000	80,000	40,000
Installation	60,000	60,000	90,000	60,000	60,000	100,000	60,000	30,000
Total Investment	288,000	288,000	356,500	320,000	333,000	560,500	419,000	228,000
$/KW	480	960	594	640	416	1,121	349	320
Total Power	600KW	300KW	600KW	500KW	800KW	500KW	1200KW	600KW
Value of power/Yr @ 3¢KWH	157,680	78,840	157,680	131,400	210,240	131,400	315,360	157,680
Fuel Savings(Loss)	10%	5%	10%	7%	-10%	7%	-25%	-10%
Value of fuel Savings/Yr @ $2/MBTU	87,600	43,800	87,600	61,320	(-87,600)	61,320	(-219,000)	(-87,600)
Total Value	245,280	122,640	245,280	192,720	122,640	192,720	96,360	70,000
Capital/Value(SimPayback)	1.174	2.348	1.453	1.660	2.715	2.908	4.350	3.25
Bbl/oil saved/year	15,200	7,600	15,200	11,800	5,240	11,800	1,423	2,320

Fuel Burning Efficiency Improvements

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New England College
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Aug. 13 - 18, 1978

Fuel Burning Efficiency Improvements

A specialized sensor has been developed to measure portions of a flame's spectra that is related to its air/fuel ratio. One Spectral Flame Analyzer is mounted on each burner of a multi-burner furnace. A closed loop control system adjusts each burner to the same air/fuel ratio.

A second sensor analyzes the flue gases and adjusts the overall excess air to a minimum value to maintain the "end point of the flame's chemical reaction" to a constant composition.

Boiler fuel saving of 2% to 4.5% have been achieved. This savings will pay back the capital investment in one to three years.

Introduction

The technical community and press are replete with dissertations, technical studies, articles, symposia, workshops, etc., on increasing fuel burning efficiency. Why another one? The answer lies with the results. In the summer of 1978 the United States is continuing to increase its oil imports over the pre-embargo levels. In the summer of 1978 the country which prides itself upon being the technology leaders - the first to put a man on the moon - is operating the majority of its industrial boilers less efficiently than any other major industrialized country in the World. The technology exists, has been demonstrated in many installations, is proven, and is available. Why hasn't it been implemented? The U. S. has always had an energy surplus. Most industrial boilers, heaters, etc., have been designed to operate with a minimum amount of human supervision, maintenance, and care. The relative availability of fossil fuels and the KISS approach (keep it simple, stupid), have resulted in the deliberate design of fuel burning systems that are both wasteful of fuel, and expensive and/or difficult to convert to the more internationally accepted control procedures.

This paper discusses an automated system which will retrofit to most newer industrial boilers and heaters, which in conjunction with possibly a few additional sensors and control hardware will typically allow continuous operation of natural gas and/or oil-fired units at less than 1% excess oxygen, and coal-fired units at less than 2% excess oxygen.

Burner Performance Characteristics

Fuel burner designers, in the historical past, up to about 1973 were primarily concerned with getting good turbulent mixing in the burner and good air fuel distribution. The high turbulence was to provide rapid and complete combustion of the fuel. The distribution system was designed so that the air flow from one burner to another was approximately ± 5% from the average air flow to all the burners. The fuel distribution system was designed to be approximately the same. Gas burners could readily achieve ± 5% distribution. The RMS distribution of these air and fuel tolerances provided 7% air flow RMS and 7% fuel flow RMS under ideal conditions. For this reason, most burner manufacturers advocated 14 to 15% excess air on gas, 17 to 18% excess air on oil, and 18 to 22% excess air on coal.

As burner degradation occurs, such as tip wear, coal nozzle wear and warpage, pulverizer classifier wear, and other general deterioration in the fuel system, the tolerances are increased by an additional 8 to 10%. The use of conventional air flow/fuel flow metering required additional increases in excess air so that some of the burners would not operate with high combustibles. An oxygen analyzer, measuring the oxygen in the flue gas,is unable to identify distribution errors in the fuel and air system or identify leaks in the furnace casing and duct work on a balanced draft firing system.

The highly turbulent rapid combustion burners are prolific generators of NO_x. Rapid combustion elevates flame temperatures sufficiently to generate large quantities of NO and NO_x. New burner designs now divide the burner air supply into several zones. These zones provide rapid mixing of primary air with the fuel, but in such a quantity that the primary air is inadequate to cause complete combustion. Therefore, the flame temperature is considerably lowered. Secondary air then mixes with the partially burned fuel further out in the furnace to complete the combustion reaction.

The flame becomes very long because of the slow chemical reaction of the air and fuel. A great many newer systems also require tertiary or over-fire air to be added in the upper part of the furnace to assure complete combustion of the fuel. It is not uncommon for the O_2 analyzer to be indicating 3 or 4% oxygen, equivalent to 17 to 22% excess air, and have some smoking on an oil fired furnace. There is no visual method by which the operator can inspect each burner and determine its air fuel ratio. A tool is needed to automatically and continuously measure each burner's air fuel ratio so that the burners may be balanced to provide the lowest excess air for complete combustion.

Spectral Flame Analyzer

An optical sensor has been designed to mount on the burner windbox to analyze certain constituents of a flame's chemical reaction, to identify the air/fuel ratio of the burning fuel. The instrument mounts on the burner front and analyzes the flame chemistry in-situ. The burner air/fuel ratio is adjusted so that the individual sensor output is identical to the mathematical average of all burners in a given firing group. The firing group is usually a horizontal row of burners. This technique permits the biasing of one row of burners differently from another row. In this way, the NO_x can be greatly reduced by having differing air/fuel ratios on different rows of burners

The Spectral Flame Analyzer is an emission spectroscopic device, using the flame as the source, and measuring on the same detector, sequentially, two separate wavelengths. The first wavelength (I) corresponds to a chemical constituent in the flame, and is a variant with respect to air/fuel ratio. The other wavelength (I_o) is a nearby blackbody emission reference wavelength, which is an invariant with respect to air/fuel ratio. The mathematical quotient of the variant wavelength (I) divided by the invariant wavelength (I_o) provides a direct measurement of the flame chemistry.

This analytical technique provides a specific flame measurement, and is independent of intensity changes due to light absorption, scattering, dirty optics, etc. It also provides excellent discrimination against other non-flame emission sources, such as glowing boiler tubes, refractories, etc. Finally, by sighting forward along the burner tip, the sensors operation is independent of flame position and displacement off of the burner tip. The ratio of I to I_o provides an intensity insensitive direct flame measurement. The only factor affecting the ratio is the strength of the emitting species.

The operator can now compare one burner to another in the same row and compare groups of burners of one row with groups of burners in another row.

The control system can be executed with analog or digital control loops. Small systems are more readily executed with analog control. Larger systems with eight or more burners can be executed more readily with digital control systems.

Either system permits the use of flame shaping and row biasing in order to achieve low NO_x without having alternated reducing and oxidizing conditions on the tube wall.

Boiler Excess Air Control

With the advent of low NO_x burners, NO_x control limits on furnace emissions, and other air pollution regulations, it is now clear that an oxygen analyzer alone is not adequate to fire a modern fuel burning system. The Spectral Flame Analyzer permits balancing the burners in the furnace, (1). A second sensor is required to measure and control the boiler at near ideal or stoichiometric fuel burning conditions.

When burning fuels that contain carbon, carbon monoxide is formed in very minute quantities at excess air levels just above stoichiometric. This carbon monoxide virtually disappears at only a few percent of excess air in the flame, but increases very rapidly when there is insufficient air to react all of the fuel. The measurement of carbon monoxide in exceedingly small quantities (0 to 300 ppm, or 0.03% CO) provides an ideal measurement of the combustion end point, allowing control at or near stoichiometric conditions, (2).

A CO analyzer can control boiler excess air to very nearly stoichiometric conditions in the actual flame regardless of the excess air that is added by over-fire air, burners out of service (BOS), air infiltration through leaks in balanced draft furnaces and other undefined causes of air infiltration. The CO analyzer described here is an in-situ optical instrument that measures CO with a one second time constant, measuring across the entire duct or stack and integrating the entire stack reading, (3). No sample is removed or manipulated in any way. Only a light beam of infrared energy passes through the flue gases.

System Diagnostics

The CO analyzer is ideal for automatic feedback control. The CO analyzer can trim the measured air flow to the combustion control system. This trimming action can cause the measured air flow reading to increase or decrease, thus affecting the overall furnace air fuel ratio. The use of the oxygen analyzer or an air flow meter in conjunction with the CO analyzer can provide additional valuable information to the operator. Maintaining a constant CO level and comparing the excess air that is leaving the furnace permits the operator to analyze the condition of his fuel burning system on a continuous basis, (4).

The CO analyzer provides a number of additional benefits which are not immediately obvious. If a furnace upset occurs, such as an oil burner plugging, or a hang up in a coal feeder which would affect a large number of burners, the total air flow/and total fuel flow measurement may not change to the furnace, but the furnace conditions would be dramatically upset. If, for example, we had a four burner boiler and one oil tip plugged, then the total fuel control would put 4/3 fuel through each of the remaining three burners and zero fuel through the fourth burner. However, each burner would have its normal air flow through four burners. High O_2 would occur from the unreacted oxygen of the burner with no fuel, and simultaneously high combustibles would occur from the three burners having 33% more fuel than air through them. Under automatic oxygen control, the air flow would be reduced and the smoking and unburned fuel reaction condition would worsen. Under automatic CO control, the system would add air or cut back fuel, depending on the combustion control arrangement, to achieve a constant level of CO again. The fact that the CO measurement only requires a one second time response enables the CO control system to immediately correct the combustion control before a major upset has occurred. A number of these CO systems are on automatic control. The operators praise them highly. The CO controls have prevented accidental boiler trips under the conditions just described.

In this situation, the Spectral Flame Analyzer and its associated control system would recognize the loss of fuel in the one burner and the excess fuel in the others. The Spectral Flame Analyzer would take corrective action, in the proper direction, to reestablish uniform air/fuel ratio conditions to the burner group involved. This rapid correction of burner conditions also aids in preventing unstable burner flames and false burner trips from occurring as they do with many flame scanning systems.

Each Spectral Flame Analyzer has an indicating meter to show the operator the output of each burner. Each meter has an adjustable high and low alarm set point on the meter front. The operator can adjusted the alarm points for each burner individually.

He can get an early warning of burner plugging, air register malfunction, coal hangups and other operating conditions which are not visually observable from the normal panel board instruments or by peeking into the furnace. The normal operating limits imposed on the Spectral Flame Analyzer system is a correction of ± 10% of the air or fuel to a given burner. This narrow limitation is intentional. If a burner deviates by more than 10%, then it is important that the operator be alerted to that fact and that maintenance of the burner be performed. This early warning can frequently prevent catastrophic failures which would occur because the failure could continue undetected.

Economics

The SAFE system payback is greatly dependent upon the particular furnace design and how well it is being operated. Large boilers with superheated steam and steam temperature controls that operate with 25% excess air can expect to save up to 4% of their fuel. Smaller furnaces with little heat recovery will save about 2% of their fuel. They will have an appreciable energy savings in their fan power, a reduction in maintenance and other benefits.

The following is an example of a small industrial boiler's potential fuel savings from reduced excess air. The savings will be in a reduced dry flue gas loss.

Boiler parameters:

Steam flow = 300,000 pounds/hour, 650 PSIG, 750°F

Net Btu/pound of steam = $\underset{\text{(stm)}}{1375.6} - \underset{\text{(f.w.)}}{218.8}$ = 1156.8 Btu/pound

Flue gas temp., Ts = 600°F

Pounds air/10^6Btu, Pa = 755 for #6 oil

Ambient air temp., ta = 60°F

Pounds fuel/10^6Btu, Pf = 55 for #6 oil

Boiler efficiency = 80%

Pounds $H_2O/10^6$Btu, Pw = 52 for #6 oil (burning H_2 in fuel)

Fuel = #6 oil at \$2.10/10^6Btu

Final O_2 w/CO control = .5%

Efficiency Gained

The flue dry gas loss (DG) is computed from the following equation:

$$\%DG\ \text{loss} = 24.10^{-6} \left[Pa \left(\frac{\%O_2}{21-\%O_2} + 1 \right) + (Pf - Pw) \right] (t_s - t_a)$$

By computing the dry gas loss at high excess air, and then again at low excess air, the difference will be the heat energy used.

$$\underset{\text{(at 4.5\% } O_2)}{\%DG\ \text{loss}} = 24.10^{-6} \left[755 \left(\frac{4.5}{21-4.5} + 1 \right) + (55-53) \right] (600-60) = 12.49\%$$

$$\underset{\text{(at 0.5\% } O_2)}{\%DG\ \text{loss}} = 24.10^{-6} \left[755 \left(\frac{0.5}{21-0.5} + 1 \right) + (55-53) \right] (600-60) = 10.06\%$$

Reduced % DG loss = 12.49% - 10.06% = 2.43% fuel saved by not heating the excess air. This excess air did not enhance the combustion chemical reaction, it only sent heat up the stack.

$ Saved

The hourly fuel cost for this boiler is:

$$\text{fuel \#/hour} = \frac{(300{,}000\ \text{\#/hr})\ (1156.8\ \text{Btu/\#})(\$2.10/10^{6}\text{Btu})}{.80\ \text{efficiency}} = \$910.98/\text{hr}$$

$$\$\ \text{saved/hour} = (\$910.98)\left(\frac{2.43\%}{100\%}\right) = \$22.14/\text{hour}$$

If the boiler operates 340 days per year at this load, the annual fuel savings will be $180,636/year.

The fan power saved will be approximately 15%. If the FD fan is 500 Hp and the electricity cost is $.035/KWH, the annual electric savings will be $22,778. The fuel and electricity savings will be in excess of $203,000/year. The added maintenance for the instrument will be 20% of a man year. It will require an instrument technician level person. This typical cost, including overhead, is about $13,000/year.

The net tangible savings will be in excess of $190,000/year. There will be other intangible savings such as reduced maintenance of boiler, burners, and auxiliary machinery through the early detection of off-normal operating conditions.

Conclusion

A system has been developed which utilizes previously developed sensor technology for direct on-line measurement to control individual burner air/fuel ratios. The use of these Spectral Flame Analyzers in conjunction with a sensitive, fast-response, long path averaging CO analyzer allows control of fuel burning processes at near stoichiometric levels. The instrumentation has been developed to be retrofittable to existing installations to provide significant and immediate paybacks. The next step is the installation and operation in a large number of facilities committed to fuel conservation and improved operations.

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FUEL CONSERVATION USING BALANCED BURNER CONTROL AND LOW EXCESS AIR FIRING

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ABSTRACT

The application of low excess air (LEA) firing to industrial combustion equipment holds the potential for both reducing fuel consumption and the combustion-associated pollution emissions of NO_x and SO_3. The utilization of LEA firing by the U.S. industrial community in the past has been limited because of the availability of cheap fuel, possible problems associated with unbalanced burners, flame stability and high labor input. Recently, advanced instrumentation has become available to automate low excess air firing in industrial and other combustion equipment. The instrumentation is called the Spectral Fuel Conservation System (SFCS) and is manufactured by Environmental Data Corporation. The SFCS is designed to balance burners in multiburner installations thereby making feasible the reduction of excess air to lower workable levels than would be practical with unbalanced burners. The lower the practical level of excess air the greater the fuel savings which can result. The U.S. Department of Energy has recognized the potential value of this type of system and is now sponsoring an evaluation, test and demonstration of the system. A multielement test program has been formulated to determine the efficacy of the SFCS and its applicability to the industrial environment. The results of this test may show that the industrial community will have available for its use instrumentation for simultaneously reducing fuel consumption and pollution emissions.

The widespread, nationwide application of low excess air firing to America's larger boilers, such as those used in industrial applications, can result in the saving of significant quantities of fuel. The greater the reduction in excess air, the greater the quantity of fuel saved. The amount which can be saved is bracketed by (1) the boiler and burner design which will determine the minimum level of excess air which can be utilized and (2) the customary excess air level utilized by the boiler operator. Subject to these considerations, fuel savings in the range of 1 to 2% are possible and even greater savings can be achieved. For the boiler operator, these savings translate into lower operating costs and increased profits.

Traditionally, the use of low excess air firing has not been widespread in the U.S. There are a number of fundamental reasons for this. Perhaps paramount has been the past availability of cheap fuel, which did not encourage the efficient use of energy and the more sophisticated control systems required to achieve the efficiency gain associated with low excess air operation. Another factor has been that flame stability is generally higher with high levels of excess air. In fact, burners designed to operate with large levels of excess air have in the past been sold for their high degree of flame stability. This is a significant factor in industrial environments where minimum operator attention to the combustion process is desirable. Further, at low excess air levels, a minor increase in fuel flow or a decrease in air flow could result in smoking or even in potentially dangerous flame instability. Avoiding these dual elements of emissions compliance and safety could also require a high level of operator attention. As fuel prices increase and saving fuel becomes increasingly important, the use of fuel saving techniques such as low excess air firing become increasingly desirable.

Although not widely used in the U.S., low excess air firing has been utilized in Great Britain to reduce the "acid smuts" emissions associated with the combustion of fuels containing high levels of sulfur. On combustion, the sulfur in fuels is converted to SO_2 and to a lesser extent to SO_3. In the presence of moisture, SO_3 converts to sulfuric acid. The latter associates with fly ash emissions which fall within a several mile radius of the source and thus have the potential to cause extensive property damage. Minimizing excess air reduces the conversion of SO_2 to SO_3 and thereby acid smuts.

Low excess air firing is also used in Japan to reduce NO_x emissions, since low excess air can also serve to minimize the formation of NO_x as well as the formation of SO_3. In Japan, the labor situation is such that an operator can be assigned to each boiler to not only monitor and control low excess air operation but also, in a multiburner installation, to balance burners to the extent permitted by the boiler's flexibility and the operator's skill.

An instrumentation system designed to automate low excess air firing-balanced burner operation on industrial boilers and other combustion equipment is also being described at this morning's session. The boiler control system designed to achieve these goals is manufactured by Environmental Data Corporation, EDC, and is termed the Spectral Fuel Conservation System (SFCS). It is also called by EDC the SAFE (for Spectral Analysis Flame Efficiency) system. Utilization of this system is designed to result in

- reduced fuel consumption
- lowered emissions of NO_x and SO_3
- reduced back end corrosion
- minimal routine boiler operator attention

The U.S. Department of Energy has recognized the need to reduce industrial fuel consumption and is sponsoring a demonstration, evaluation and test of the EDC system on an operating industrial boiler. In the test program, sponsored by DOE under Contract No. EY-76-C-02-2913, Exxon Research and Engineering Company, ER&E, is prime contractor and is performing overall program management as well as test program design and evaluation of the data to be obtained. Environmental Data Corporation has constructed, installed and is operating and maintaining the SFCS and aiding in analysis of the data.

The SFCS is designed to operate as a trim on an existing industrial boiler's control system. The basic elements of the SFCS are the Spectral Flame Analyzer (SFA), an in-stack pollution monitor, control hardware and logic circuitry. Two control loops are used in the system, one to balance the burners, the other to control the overall excess air. In the first, one SFA directly sights each flame and determines its stoichiometry. In a multiburner installation, the SFAs are used to keep the fuel/air ratio at each burner identical. In essence, this loop of the system is designed to make a multiburner installation perform like a single burner installation. For this test and demonstration program, burner control starts with the determination of the fuel/air ratio of each of the four burners of an industrial boiler by the SFA. An average fuel/air ratio is then calculated from these measurements by analog computer. Should one burner differ from the average, the fuel flow to that burner will be adjusted to make its stoichiometry identical to the others. This action produces a change in the average and the process is repeated. Fuel/air ratio measurement occurs continuously and changes in fuel flow are accomplished on a real time basis.

The CO signal from the in-stack emissions monitor is used to control the overall excess air level in the boiler. The CO concentration varies inversely as a function of excess air level. However, at a very low excess air level, the CO concentration increases rapidly due to incomplete combustion of the fuel. The logic circuitry of the SFCS is designed to hold the overall excess air level at a minimum without approaching critically CO levels which are symptomatic of flame instability and which may be regarded as being environmentally undesirable.

If the excess air level were reduced without individual fuel/air ratio control on each burner, the burners typically would not be firing under identical stoichiometric conditions. Some might be fuel rich and would be sources of pollution; others might be fuel lean and operating with a considerable excess of air and consequently wasting fuel. With the SFCS, all burners will be firing in unison. Thus, excess O_2 should be capable of being reduced to the 1% level or less with resultant fuel savings. In addition to reduced fuel usage, pollutant emissions of NO_x and SO_3 should also be reduced. Reduced levels of SO_3 should also result in lowered back end corrosion.

The EDC instrumentation is based on operating principles which are completely general and this system should be applicable to the combustion of virtually any hydrocarbon fuel in any combustion device where the flame can be properly sighted by the SFAs.

This DOE-sponsored test, evaluation and demonstration is being conducted on a dual fuel (oil and gas) four burner Babcock and Wilcox industrial boiler which has a maximum generating capacity of 300,000 pounds of steam per hour. The burners are placed in a rectangular pattern on the burner front. The boiler is owned by Kerr-McGee Chemical Corporation and is located in Trona, California. We have received excellent cooperation from Kerr-McGee in conducting this program. The test program can be divided into five elements:

1. Basic Test
2. Automatic Control Test
3. Biased Firing Evaluation
4. Flame Presence Test
5. Long Term Test

The objectives of the basic test program are to (1) measure fuel savings, (2) evaluate the effectiveness of the SFAs at equalizing air/fuel ratios under different load conditions and (3) determine the parameters for automatic operation of the boiler by the SFCS. To accomplish these objectives, data are being obtained at (a) three different excess air levels, (b) three different loads and (3) with and without the SFA control loop in operation. The data being obtained include:

- Fuel consumption
- Flame stoichiometry
- Emissions of CO, NO, CO_2, CO, opacity
- Excess O_2

During the automatic control test, the boiler will operate under the closed loop control of the SFCS using the parameters established during the basic test. This test phase will serve to evaluate the automatic control of boiler operations and to identify control problems. Load changes typical of industrial boiler operation will be programmed to evaluate the ability of the SFCS to control combustion during changes in load.

The biased firing evaluation will determine the ability of the SFCS to control off-stoichiometric combustion used to reduce NO_x levels. In this case, the two lower burners will operate at an identical, but fuel rich fuel/air ratio and the upper burners will be equalized at a full lean ratio. With the proper overall burner "pattern", the NO_x emissions of many boilers can be reduced even further than by the application of low excess air firing while at the same time reaiming the low excess air firing efficacy gain. The gases in the test boiler flow from front to back rather than upward as is the case with many boilers and as a consequence little reduction in NO_x is likely with this specific boiler.

The flame presence test will evaluate the ability of the SFA signal to serve as a flame presence monitor. If the test is successful and after appropriate approval has been obtained from insurance associations, the SFAs may be certified to replace existing flame proving devices.

For the long term test, the boiler will operate for six months under the closed loop control of the SFCS. During this period, by means of mechanical, electronic and performance checks, any change in the system or its ability to properly and effectively control combustion will be determined. This will measure any deterioration occurring to the system attributable to continuous operation and to the severe industrial environment where the test is being run.

The test and demonstration program described here is designed to determine the efficacy of the SFCS and its compatibility with the industrial environment. Assuming the test is successful, industrial combustion facilities will have available a new technique for reducing fuel consumption. Possible future programs to further evaluate this system in an industrial environment include the application of SFCS to process furnaces and to coal fired boilers. These represent significantly different categories of combustion equipment, and can represent the application of this form of energy conservation technology to the largest industrial users of fuel.

78-9E9-HPGDR-P1

Unrestricted
G. W. W
Division Manager
7/20/78
Date

ELECTRIC HEAT PUMPS FOR GRAIN DRYING

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O. C. Doering, G. H. Foster, M. R. Okos
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ELECTRIC HEAT PUMPS FOR GRAIN DRYING

M. R. Hogan
Heat Transfer and Fluid Dynamics

O. C. Doering, G. H. Foster, M. R. Okos
Purdue University

ABSTRACT

Tight supplies of the convenience fossil fuels and the urgent need for energy conservation have prompted investigations into efficient, reliable, alternate heat sources for grain drying. The heat pump is an especially viable alternative source because it delivers more energy as heat than it consumes. In addition, it utilizes one of the most reliable energy forms available to the farmer--electricity.

This paper discusses the energy requirements of agriculture with respect to grain drying. Current drying methods as well as potential heat pump systems are reviewed. Recent development work and test results from a prototype heat pump dryer are presented, and a review of current heat pump drying research and development is given.

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ELECTRIC HEAT PUMPS FOR GRAIN DRYING

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Heat Transfer and Fluid Dynamics

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SECTION 1.0

INTRODUCTION

Field-shelled corn requires drying before storage to prevent spoilage due to mold growth. Present forced-convection grain drying methods rely on liquefied petroleum gas and, in some instances, natural gas. The need for energy conservation and tight supplies, aggravated by the seasonal nature of grain drying, have prompted investigations into efficient, dependable, alternate heat sources for grain dryers. Low-temperature dryers utilizing joule heating are increasing in popularity because of the inherent efficiency of low-temperature drying, and the availability and supply stability enjoyed by electrical energy. The heat pump is a device which can utilize electrical energy but, for the same heat delivery rate, at 15% to 25% the power consumption of an I^2R heater.

This paper deals with the unique role energy plays in corn drying and some of the present LPG supply difficulties. Current grain drying practices are reviewed and a brief description is given of potential heat pump grain drying applications. Recent Westinghouse and Purdue University research and heat pump dryer development is discussed, as is current Department of Energy-sponsored work.

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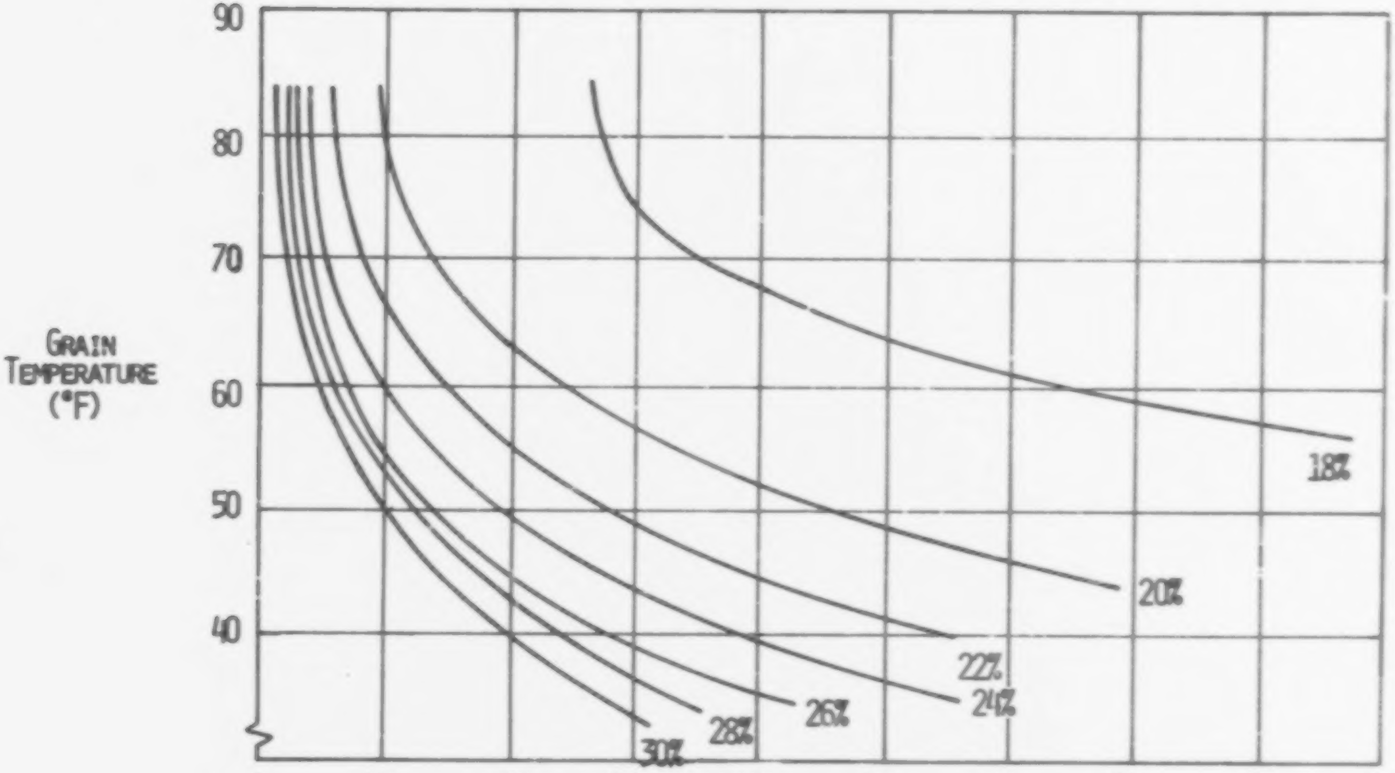

FIGURE 2-1. ALLOWABLE STORAGE TIME FOR SHELLED CORN. DATA FROM THE U.S. DEPARTMENT OF AGRICULTURE GRAIN STORAGE RESEARCH LABORATORY, AMES, IOWA.

SECTION 2.0
ENERGY AND GRAIN DRYING

Heated-air, forced-convection grain drying is a relatively recent development in agriculture and is a result of shifts in grain handling and marketing practices brought on by large increases in grain production. As yields doubled over the 1950's and 60's from increased application of nitrogen fertilizer and hybrid seed technology, farmers experienced severe handling problems with ear corn, normally dried over the fall and winter months in open air cribs. Given the spaces between ears, natural airflow was usually sufficient to dry the corn so that spoilage would not occur.

The immediate advantage of shelled corn was that it was a free-flowing product of substantially less bulk than ear corn. Cobs did not have to be transported from the field, and the second operation of removing the corn grain from the cob was accomplished simultaneously with the harvesting. In a one-step operation, shelled corn systems provided the farmer with a compact, free-flowing product in its final form for marketing.

Shelled corn, unlike ear corn, however, could not be stored without drying. Figure 2-1 shows the moisture-spoilage rate relationship. For a 24% moisture content, a typical harvest value, only ten days can elapse before spoilage occurs if the grain temperature is 60°F. Nevertheless, the factors of handling, labor requirements, and marketing convenience resulted in a rapid adoption of shelled corn technology, in spite of the energy costs associated with drying.

If corn is allowed to dry while standing in the field (which it will do with little spoilage under average weather conditions), more than enough is lost to pests or lodging than is needed to pay for the drying fuel. Recent studies show that LP gas would have to cost $2.80 a gallon in order for it to be worthwhile for the farmer to let corn dry naturally while standing in the field[1]. Thus, timeliness in harvesting is critical to the yield, and timeliness in drying is critical to the preservation of the crop. The farmer's optimum strategy depends on trading off a high-yield, early harvest with its concommitant drying costs against a later, drier harvest with larger field losses.

One of the major concerns with respect to grain drying has been the use of premium fuels, LP gas and natural gas. Most farms are located away from natural gas lines, so the bulk of on-farm grain drying depends upon LP gas. These fuels are most commonly used because exhaust gases can pass through the grain for the greatest efficiency of fuel use without contaminating the grain. As long as LPG costs less than oil on a heat-delivered basis, there is certainly no way one would move to the more expensive oil.

We face some potentially critical problems with LP gas. These relate to both supply and distribution, with distribution probably the most critical. Liquefied petroleum gas was first a by-product of the petroleum refining process, and a market was developed on that basis. Its easy liquefaction made it a premium portable fuel, and demand soon began to outstrip the supply available from petroleum refining. The growth of the natural gas industry made butane and propane available from this source, and now two-thirds of the LP gas used in the United States comes from natural gas production, and only one-third from petroleum refining. As natural gas becomes less available, so too will LPG.

Transportation and distribution to the rural areas of the midwest and north central states also pose increasing problems. Pipeline capacity is increasingly being utilized by natural gas and oil, with less capacity available for LP gas. The safety aspect, if nothing else, raises serious questions as to whether rail transportation can be depended upon for LP gas delivery. Beginning with the fall of 1972, every late wet harvest season has resulted in the overlap of grain drying with the start of the home heating season and spot shortages of LP gas occurred for both grain drying and home heating. This is the reason LP gas came under federal regulation in mid-1973, prior to the Arab oil embargo.

Users competing for LPG do not necessarily have the same market leverage. As much as half the LP gas supply is used in petrochemical production, much of it located in the Gulf and southwestern states near the LP gas producers. The reamining supply is sent to the Midwest and other areas of the country, where it is sought by small businesses, home owners, and farmers. These three groups are in competition for the supply remaining after the needs of the petrochemical industry have been satisfied. All three require a decentralized transportation and distribution network which can only meet peak demands by maintaining excess capacity. The agricultural demand varies so much from year to year that it is increasingly uneconomical for LP gas dealers and distributors to maintain the excess supply and delivery capacity to meet peak years.

The reliability of drying fuels is crucial to the farmer. This fosters a unique attitude toward risk. There is every incentive for substantial investment since the product has already accumulated a total cost of nearly \$2.00/bu and a market value of 2.20 to 2.30\$/bu. Thus, the 7-10¢/bu drying cost is a small amount to pay if it will preserve his investment. Thus, there is concern among farmers about the reliability of fuel supplies for drying; therefore, reliability takes priority over price. This is one of the main reasons why farmers are increasingly interested in alternative drying systems, particularly those that might not be subject to the potential for increasingly unreliable LP gas supplies. In addition, there is the national concern for energy conservation. Thus, any drying alternative must satisfy both criteria, energy conservation and supply reliability. A system which is only energy conservative will not be accepted by the agricultural community. It is in this context that it is worthwhile to investigate the feasibility of heat pump grain drying.

SECTION 3.0

CURRENT GRAIN DRYING PRACTICE

While artificial drying at grain elevators dates back more than 100 years, most drying on farms was by natural means until about 1960[2] in step with the advent of field shellers. From 70-85%[3] of the 1977 corn crop produced in the central United States corn belt states of Iowa, Illinois, and Indiana was dried on the farm. While much of this discussion is applicable to both grain elevators and on-the-farm drying, it is directed toward techniques utilized by the farmer.

Two rather distinct approaches to drying corn have evolved. As one might expect, the first attempts to dry shelled corn were similar to those used for drying ear corn, except that a mechanical fan replaced the wind as the driving force to move air through the more densely packed shelled corn. Heat was added to the air in order to speed drying of the more perishable shelled corn. However, this was ineffective without increasing the airflow. To increase airflow, it was necessary to use rather shallow beds compared to full storage bin depths. Thus, high-speed batch and continuous-flow dryers came on the grain drying scene. Bin drying systems persisted, however, and current efforts toward energy conservation have renewed interest in low-temperature, in-storage drying systems that make maximum use of the drying potential of natural air. Thus, we have the two systems: low-temperature, in-storage; and high-temperature, high-speed drying.

There are many variations of each of the two major types of drying systems. Low-temperature systems are used with or without added heat, and with heat from various sources--electric, LPG, natural gas, solar, etc. Drying is accomplished in full bin systems and in systems where wet corn is added

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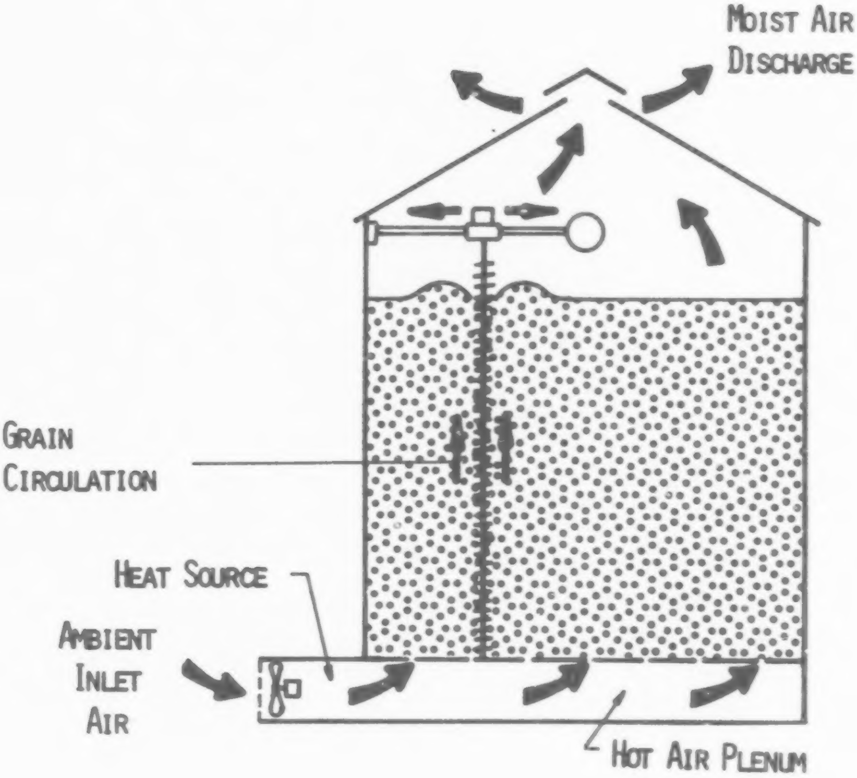
Moist Air Discharge
Grain Circulation
Heat Source
Ambient Inlet Air
Hot Air Plenum

Figure 3-1. Full Bin Low-Temperature Grain Dryer With Stirring.

in layers as drying proceeds. Low-temperature drying is characterized by deep grain beds, low airflow rates, slow but efficient drying. Drying proceeds over a period from several days to six weeks or more. The constraint on the maximum length of drying time, and consequently the minimum airflow rate required, is the perishability of the grain.

Advantages of low-temperature drying are that harvesting and filling of bins can proceed at a rate more rapid than the drying rate during good harvest weather and drying can catch up during weather unfit for field work. Low-temperature drying permits maximum use of the drying potential of natural air and is compatible with low level heat sources such as electrical resistance and solar insolation. Increased concern over past and future energy shortages has resulted in a growing market for I^2R dryers. As discussed earlier, the farmer is primarily concerned with reliability of supply, and electrical energy is the most stable energy form available to him for crop drying.

Figure 3-1 illustrates a full bin, low-temperature drying system. Ambient air is ingested by a fan, heated 5-10°F, and then forced through a perforated floor into the grain column. The drying air exits the grain near saturation, and exits the system through vents in the bin roof. Although the grain bed is often unstirred, the figure shows a typical stirring auger system. If the grain is unstirred, the top layers remain undried for a considerable period, increasing the risk of spoilage, while the lowest layers are oftentimes overdried. Stirring eliminates these risks. Figure 3-2 shows the older, layered, low-temperature system. Here grain is added as the preceding layer begins to dry appreciably. This technique suffers from overdrying at the lower levels, and a high risk of spoilage in the upper levels. This scheme is still in use but requires very careful management.

The principal disadvantage of low-temperature drying is that it is weather-dependent. It provides little reserve drying capacity, and when crop or

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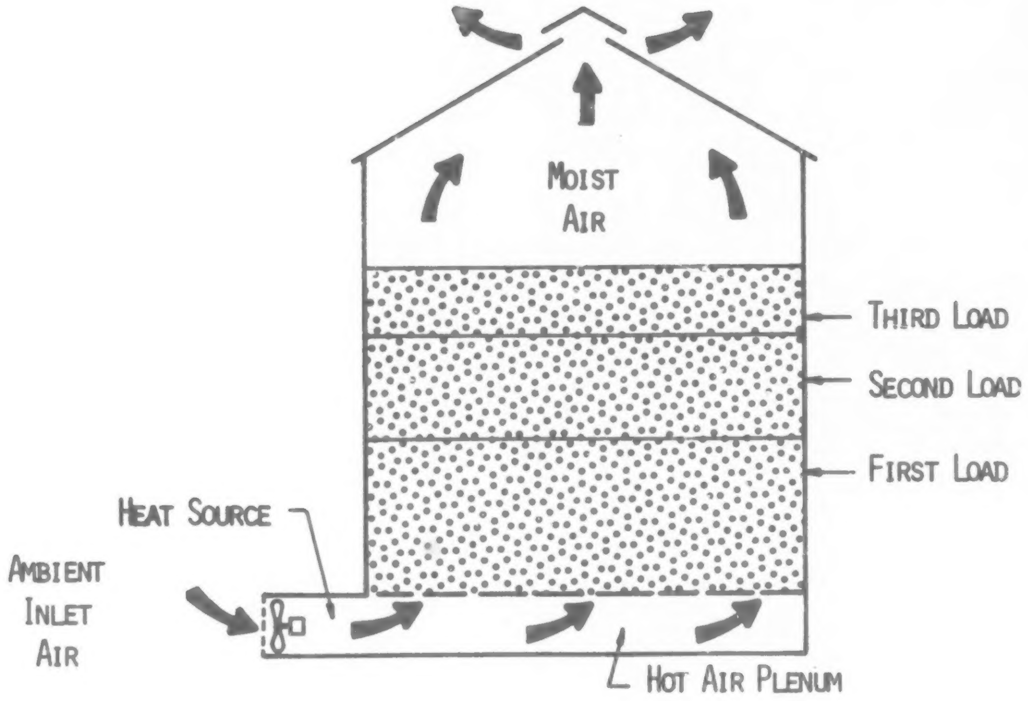
Moist Air
Third Load
Second Load
First Load
Heat Source
Ambient Inlet Air
Hot Air Plenum

Figure 3-2. Layered Bin Low-Temperature Grain Dryer.

weather conditions are unfavorable, may not provide a fail-safe drying system. The exponential increase of fan power with increased airflow limits the feasibility of increasing the drying rate significantly in full bin systems. Since drying occurs over several weeks, marketing flexibility is limited.

Containers designed for ease of filling, emptying, and holding grain in shallow layers (6-18 inches) are used for most high-temperature batch and continuous-flow dryers. Drying capacities range from 100 to 1000 bu/hr with airflow rates in the 75 to 100 cfm/bu range. As seen in Figure 3-3, the air is heated to approximately 200°F as it is forced into a plenum surrounded by a relatively thin layer of grain. The drying air passes through the porous wall into the grain bed. As the grain reaches the lower half of the dryer, tempering air cools the grain before it leaves the dryer. There are numerous variations in high-speed dryer design. Although the system shown is portable, there are permanently-fixed systems. Some dryers are constructed as a column with the grain cascading downward, counter to the heated airflow. High-temperature drying allows maximum marketing flexibility and minimizes the risk of spoilage.

Heat pump grain drying provides a number of advantages over current methods. First, it can reduce overall energy consumption. A heat pump exhibiting a heat source coefficient of performance (heat delivered [Btu/hr] divided by compressor energy consumption [Btu/hr]) of 5.0 delivers five times as much heat as is required to power the device. If an electrical generation and transmission efficiency of 33-1/3% is assumed, the heat pump delivers 66-2/3% more heat than a comparable LPG system for the same source fuel consumption. Secondly, the heat pump shifts the energy source from LPG, an increasingly unreliable fuel, to electrical energy, an historically stable energy form. In addition, the shift allows other source fuels such as coal and nuclear, which if used directly would be unacceptable for drying, to be utilized. These two advantages

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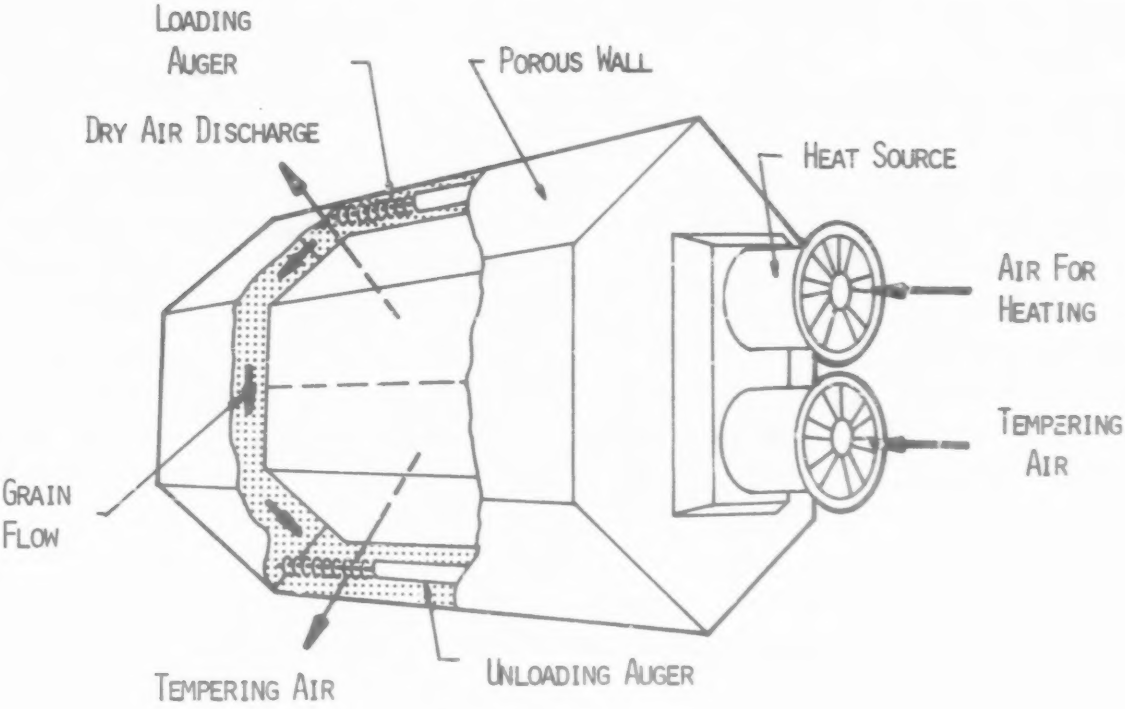
LOADING AUGER
POROUS WALL
DRY AIR DISCHARGE
HEAT SOURCE
AIR FOR HEATING
TEMPERING AIR
GRAIN FLOW
TEMPERING AIR
UNLOADING AUGER

FIGURE 3-3. HIGH-TEMPERATURE CONTINUOUS-FLOW GRAIN DRYER.

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make the heat pump a prime alternative to the increasingly questionable current drying systems.

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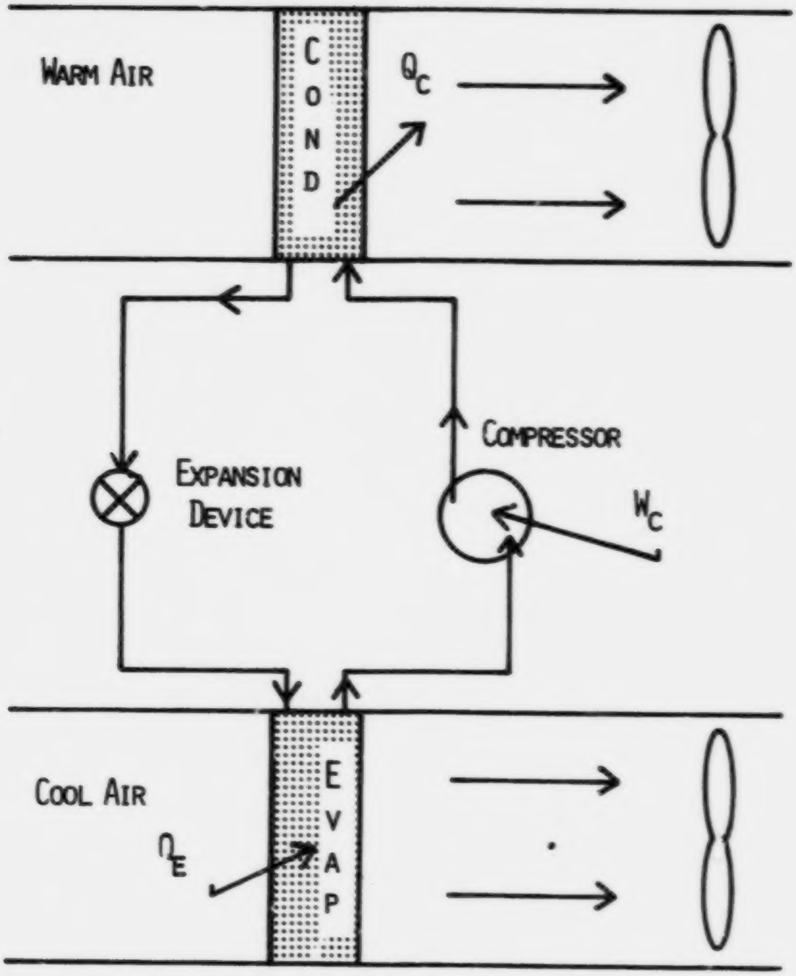
WARM AIR
COND
Q_C
EXPANSION DEVICE
COMPRESSOR
W_C
COOL AIR
EVAP
Q_E

FIGURE 4-1. AIR-TO-AIR HEAT PUMP.

SECTION 4.0

HEAT PUMP GRAIN DRYERS

Electrical heat pumping was first considered as an energy source for grain drying in the late 1940's and early 1950's. Davis[4] in 1949 studied a continuous heat pump grain dryer. Shove[5] made a similar investigation, but with a laboratory batch grain dryer in 1953. Recently, some farmers in the South have purchased residential heat pumps and modified them for low-temperature drying. However, no detailed reports on the performance of these systems are available. Heat pumps have been used to "finish" dry popcorn. In this operation, drying the corn to a level slightly above the desired moisture content is performed with conventional equipment. The corn is then finish dried with a heat pump to the desired 'popping' level.

The heat pump removes thermal energy from a low-temperature heat source and rejects thermal energy to a higher temperature heat sink. In accordance with the Second Law of Thermodynamics, this action cannot be accomplished without the expenditure of work. In the modern heat pump, the work, normally in the form of electrical energy, is required to drive the refrigerant compressor.

Figure 4-1 is a schematic of an air-to-air heat pump showing the four major components. In the evaporator, heat from the cool airstream evaporates the working fluid or refrigerant. Compressed to a higher pressure and temperature by the compressor, the gaseous fluid condenses in the refrigerant condenser, thus ejecting heat to the warm air. The expansion device maintains a pressure difference enabling the two heat exchangers to operate at different temperatures. When the heat pump is used as an air conditioner, it removes heat from a living space and

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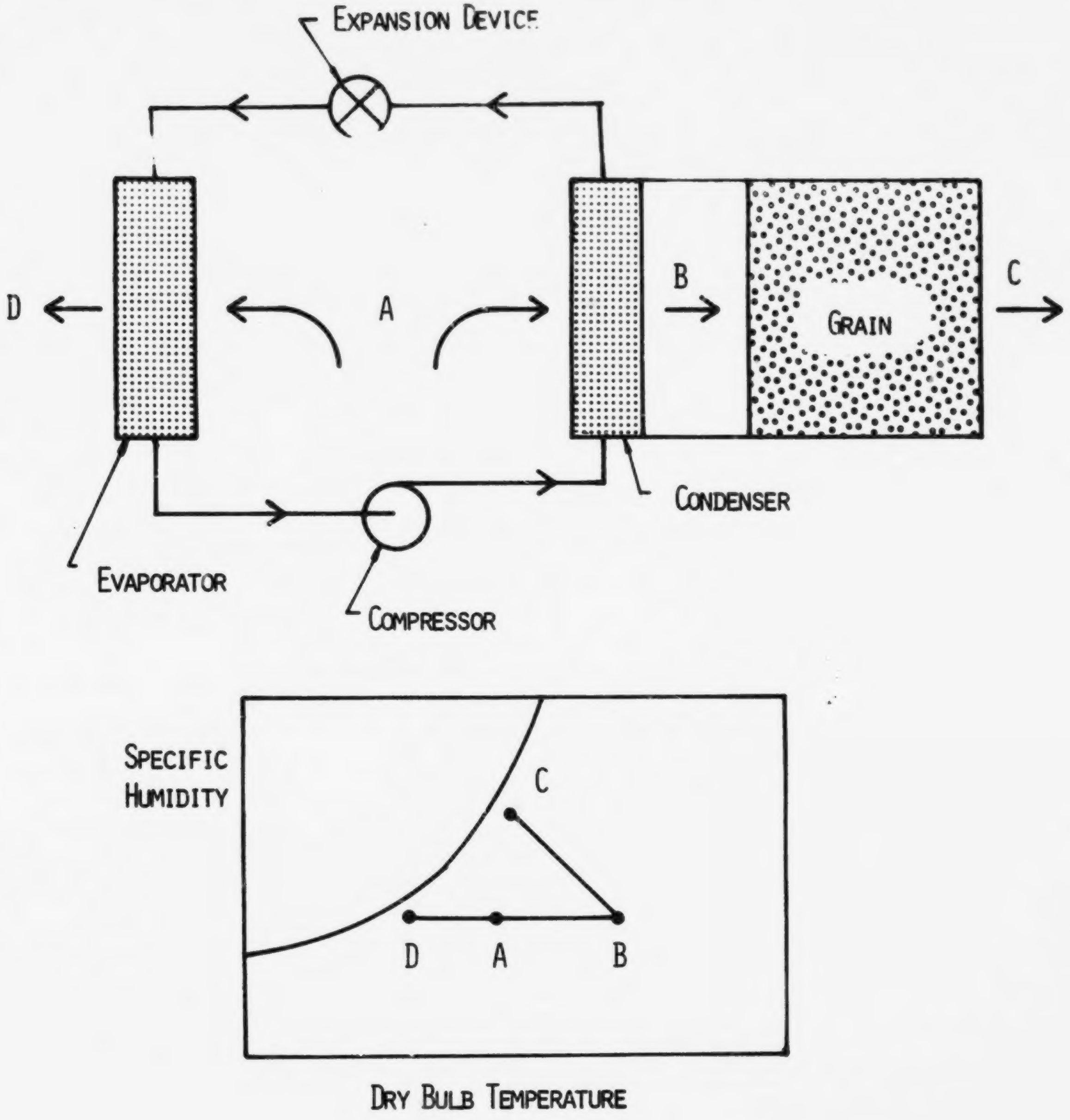
EXPANSION DEVICE
D
A
B
C
GRAIN
EVAPORATOR
COMPRESSOR
CONDENSER
SPECIFIC HUMIDITY
C
D
A
B
DRY BULB TEMPERATURE

FIGURE 4-2. OPEN-AIR-LOOP HEAT PUMP GRAIN DRYER WITH AMBIENT HEAT SOURCE AND PSYCHROMETRIC CHART.

transfers it to the warmer ambient air outside the space. When the device is used to heat a living space, heat is removed from a cool ambient exterior and delivered to the warmer living space.

The importance of the heat pump as an energy-delivering device is that it delivers more thermal energy (Q_c) than the work (W_c) required to effect the delivery. From the First Law of Thermodynamics, considering the working fluid as a system and ignoring any superficial heat losses in the connecting tubing, the heat transferred from the condenser (Q_c) is the sum of the heat transferred to the evaporator (Q_c) plus the compressor work (W_c). Thus, from the standpoint of energy conservation, the heat pump is inherently superior to joule (electrical resistance) heating in which the energy delivered as heat is equal to the electrical energy consumed.

The coefficient of performance (COP) of a heat pump is defined as the thermal energy delivered divided by the work required to bring about that delivery. Therefore, $COP = Q_c/W_c$. Oftentimes other energy inputs such as fan power are included in the W_c term in the denominator. To avoid confusion, the above definition will be designated the heat source coefficient of performance (COP_h). The system coefficient of performance, however, will be defined as $COP_s = Q_c/(W_c + W_f)$ where W_f is the fan power.

Heat pump grain dryers can be classified into two general groups: open- and closed-air systems. Open systems are once-through processes. In closed systems, the air exhausted from the dryer is partially or wholly recirculated through the heat pump and dryer.

The open-loop dryers of interest can be categorized by heat source. Figure 4-2 depicts an open-air-loop dryer with the ambient environment as the heat source. Ambient air is forced through the evaporator and cooled. The sum of the compressor work and the thermal energy extracted by the evaporator is then rejected by the condenser to the process air,

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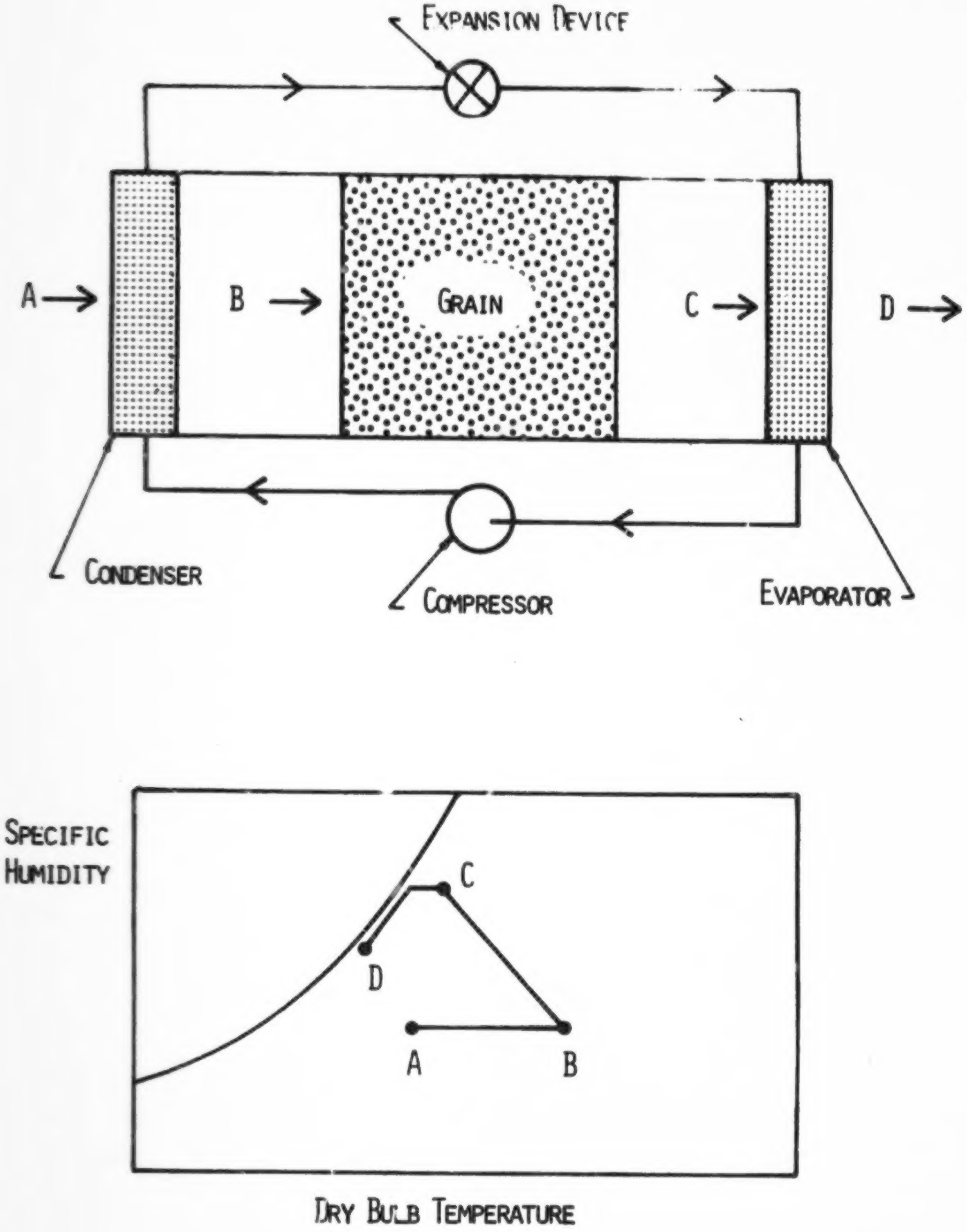
Expansion Device
A
B
Grain
C
D
Condenser
Compressor
Evaporator
Specific Humidity
C
D
A
B
Dry Bulb Temperature

Figure 4-3. Open-Air Loop Heat Pump Dryer With Dryer Effluent Heat Source And Psychrometric Chart.

which is also taken from ambient, thus conditioning it for drying. After passing through the grain column, the air is exhausted to the environment. This type of heat pumping (the local ambient environment serving as the heat source) is employed in residential systems.

Figure 4-3 illustrates an open-air system in which the grain column effluent is utilized as the heat source. As seen in the accompanying psychrometric chart, the grain effluent is usually (for typical ambient relative humidities) at a higher temperature than ambient air. In addition, it is saturated, or very close to saturation, and as a result, changes temperature less than the drier ambient air for a given change in specific enthalpy. Both characteristics provide an effective heat source temperature higher than ambient. Since the coefficient of performance is a strong function of the temperature difference between the heat source and heat sink, the heat pump COP_h is enhanced. The heat pump capacity also increases as a result of the higher gas density at the compressor inlet and the smaller pressure ratio.

Figure 4-4 is a schematic of a closed-air-loop heat pump grain dryer. In this system the grain effluent is forced through the evaporator and then returned to the condenser for reheating. Ideally no air would be vented, and only liquid water would leave the system. This is not possible for any length of time, however, because the grain mass temperature would continually increase. In actual operation, ambient air must be bled into the system to maintain the desired dryer inlet air state. This system does, however, possess several distinct advantages. Without ambient bleed, the closed-air-loop system allows a cold grain mass to reach operating condition, i.e., a condition which precludes frost formation on the evaporator heat transfer surfaces. With controlled ambient air admission, as shown in the psychrometric chart, it can maintain a system at an elevated temperature despite low ambient temperatures. This is of signal importance in preventing frost accumulation. With 100% ambient air admission the closed-air-loop is equivalent to the open-air-loop system with the dryer effluent heat source.

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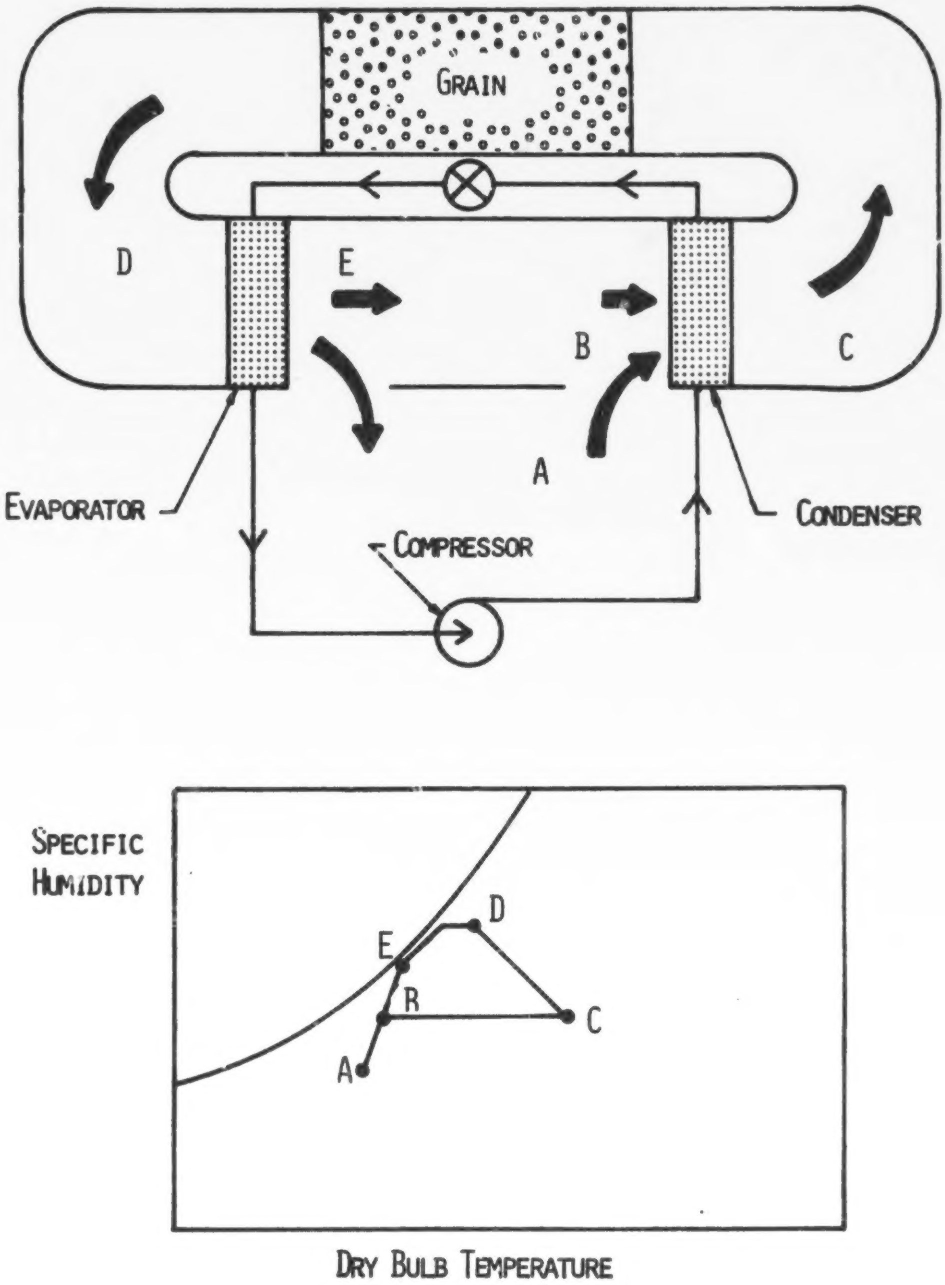
GRAIN
D
E
B
C
A
EVAPORATOR
CONDENSER
COMPRESSOR
SPECIFIC HUMIDITY
D
E
R
C
A
DRY BULB TEMPERATURE

FIGURE 4-4. CLOSED-AIR-LOOP HEAT PUMP GRAIN DRYER AND PSYCHROMETRIC CHART.

SECTION 5.0

DEVELOPMENTS IN HEAT PUMP GRAIN DRYING

The first major step toward the practical application of heat pumps for grain drying taken by the Westinghouse Fluid Systems Laboratory was in a 1974 research program sponsored by the Corporation. This program was aimed at determining the feasibility of heat pump grain dryers. As a result of this work, a computer program capable of specifying the required heat pump components for a given drying duty was developed. The code is independent of the particular drying process being studied so that any type of drying configuration may be examined. For instance, a low-temperature, open-air-loop dryer can be designed by first specifying the desired inlet air state. A grain drying simulation model can then be used to predict the outlet air state. Given these two air states, ambient conditions, maximum heat exchanger face velocities, the desired refrigerant static pressure losses, the minimum temperature difference between the air and saturated refrigerant in both heat exchangers, nondimensional compressor performance parameters, nondimensional heat exchanger geometry, and the initial number of refrigerant circuits, the code can perform a paper design of the system which will provide the necessary thermal energy exchange. Specifically, the program provides detailed heat exchanger geometry (e.g., total tube length, tube inside and outside diameter, number of rows and tubes per row, longitudinal and transverse tube spacing), the thermodynamic cycle, COP_h, air-side pressure losses, and refrigerant flow rate. In addition, the code calculates the distance in the evaporator refrigerant circuit that the number of parallel paths should be increased to avoid overly large pressure losses, and warns if frost is likely to occur on the tube surface. Since the code generates the equipment required as well as the operating conditions, it is possible to quickly and accurately determine component trade-offs that minimize total operating costs.

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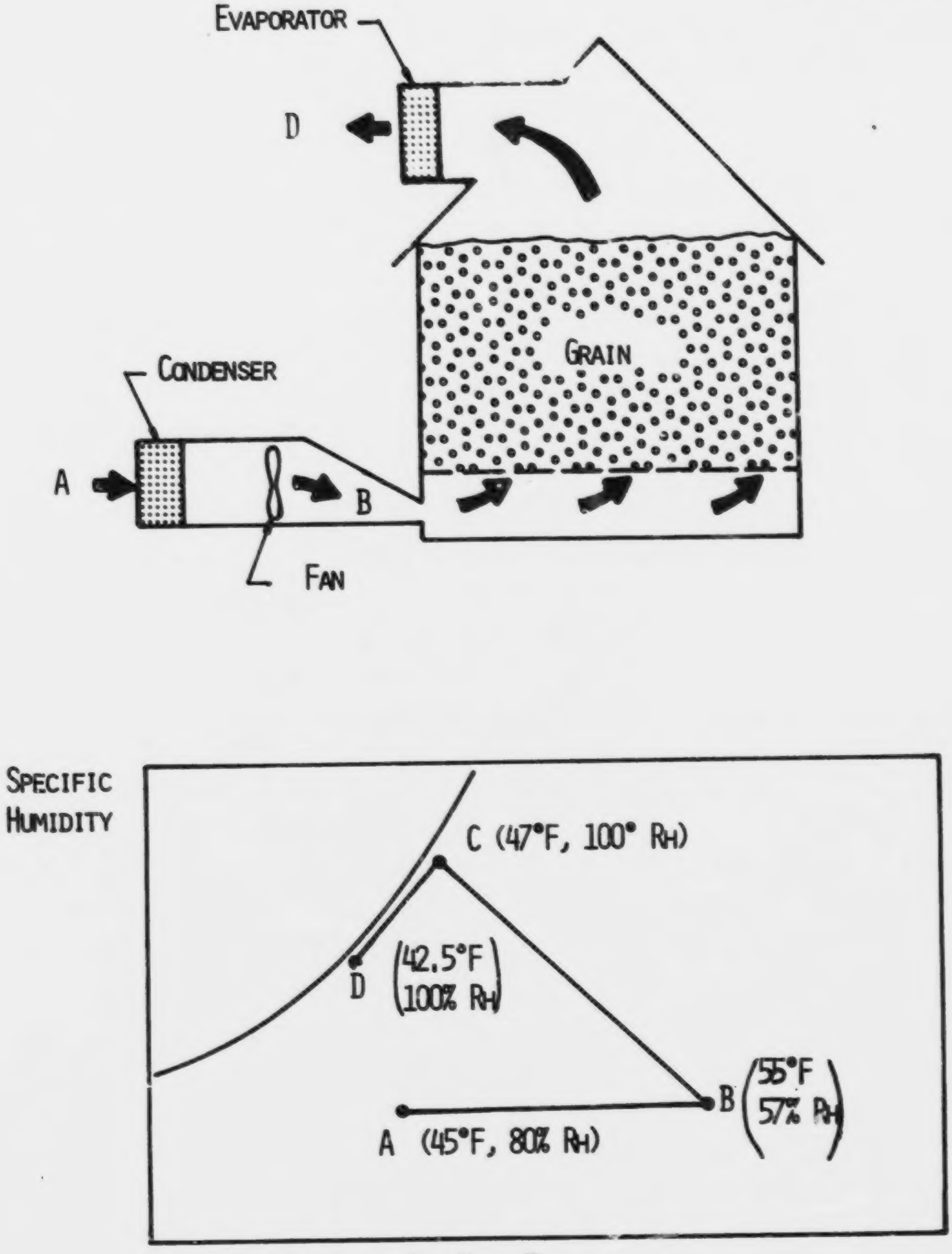
EVAPORATOR
D
GRAIN
CONDENSER
A
B
FAN
SPECIFIC HUMIDITY
C (47°F, 100° RH)
D (42.5°F 100% RH)
B (55°F 57% RH)
A (45°F, 80% RH)
DRY BULB TEMPERATURE

FIGURE 5-1. LOW-TEMPERATURE HEAT PUMP GRAIN DRYER DESIGN DUTY AND SYSTEM SCHEMATIC.

In a 1975 corporate research project. the Westinghouse Fluid Systems Laboratory and the Purdue University School of Agricultural Engineering embarked on a joint demonstration of a full-scale heat pump grain dryer. A low-temperature, open-air-loop dryer was selected for the demonstration. Figure 5-1 shows the air state points which the heat pump was required to produce. Note that steady state, as well as a saturated exit air state from the dryer volume, was assumed. The latter assumption was suggested from analytical drying simulations for wet (25% wb) corn. Process A-B represents the heating which occurs through the heat pump condenser, B-C (an isoenthalpic line) approximates the steady state drying process, and C-D approximates the energy extraction process as the grain effluent passes through the evaporator. Initially only A, B, and C were known. State D was known only after the heat pump performance was determined, but is shown in the figure and represents the results of the finalized design.

In addition to the required air states, three factors strongly influenced the heat pump design. First, the evaporator had to operate at the design ambient without frosting. Second, it was desirable to maintain the refrigerant thermodynamic cycle in the same general range as that experienced by residential heat pumps or air conditioners. Although this depressed the coefficient of performance, it was a conservative design philosophy appropriate for the prototype. Third, the airflow rate (approximately 9000 cfm) was many times larger than commonly experienced by heat exchangers used in residential units.

From the results of the code analysis, it was possible, with the aid and recommendations from the Westinghouse Central Residential Air Conditioning Division engineering staff, to select available components which were close to what was desired. Resulting from this process was a condenser consisting of 135 ft. of 3/8-inch-od copper tubing with 16 aluminum fins per inch, arranged in four parallel paths one tube row deep. The total heat transfer area was 198 ft^2 with a 28 in x 60 in face area.

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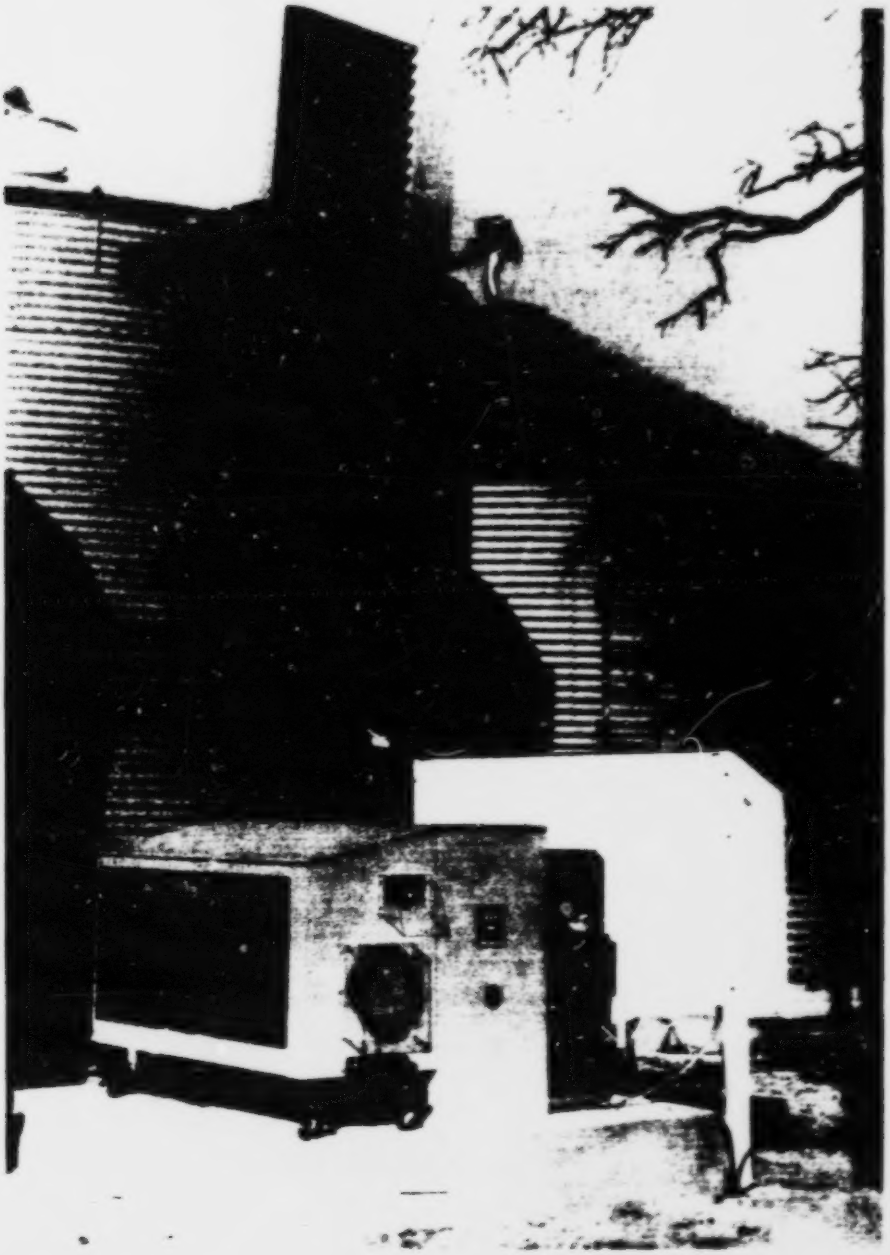


FIGURE 5-2. PROTOTYPE LOW-TEMPERATURE HEAT PUMP GRAIN DRYER.

The evaporator consisted of 525 ft of 3/8-inch-od copper tubing with 13 aluminum fins per inch, arranged in 18 parallel circuits three tube rows deep. A surface area of 692 ft^2 was available for heat transfer The face area was 44 in x 48 in. A nominal 6-3/4 ton compressor, protected by an accumulator and high and low pressure cutout, powered the system. Control of the refrigerant flow rate was achieved by a constant superheat expansion valve, a common practice in the air conditioning industry.

After construction, the heat pump was coupled to the 24-ft-diameter by 16-ft-high grain bin which served as the dryer volume (Figure 5-2). Two stirring augers were installed in the bin to provide uniform corn drying rates. Air was drawn through the heat pump condenser, situated at the intake of a 10 hp centrifugal fan shown at the bottom of Figure 5-2; forced through the perforated floor and grain column; and then through the evaporator, located on top of the grain bin, where exhaust air was utilized as the low-temperature heat source.

Since construction, the unit has successfully dried 12,000 bu of corn over three drying seasons. The drying performance (typically over two to three weeks) was closely monitored by instruments which deduced airflow rate, power input, capacity, air temperature rise, refrigerant flow rate, and the advance of the drying front. The results of these measurements showed the unit delivering 103,036 Btu/hr to the drying airstream on a 50°F day. This, plus some heat provided by the fan, provided the desired 10°F air temperature rise. The heat source coefficient of performance [COP_h] (the condenser output [Btu/hr] divided by the compressor electrical input [Btu/hr]) was determined to be 4.47, whereas the system coefficient of performance [COP_s] (the sum of the condenser and fan heat output divided by the sum of the compressor and fan electrical input) was 2.31.

It was also determined, knowing the total power consumption over the drying period and the quantity of water removed from the grain, that the heat pump system required 716 Btu as input per pound of water removed during 1975, and 760 Btu/lbm of water removed during the 1977 season. (The difference in the two measurements is due to varying annual weather conditions.) These figures for the heat pump grain dryer compare favorably with 1500 Btu/lbm, characteristic of current low-temperature electric resistance dryers, and 2500-3000 Btu/lbm for high-temperature systems.

Since approximately 6.5 lbm of water is removed in drying a bushel of corn from 25% to 15% moisture content, the heat pump energy cost per bushel is,

$$\frac{\text{716 Btu supplied}}{\text{lbm water removed}} \times \frac{\text{6.5 lbm water}}{\text{bu}} \times \frac{\text{1 kWh}}{\text{3413 Btu}} \times \frac{3.0¢}{\text{kWh}} = 4.09\ \frac{¢}{\text{bu}}$$

Because the heat pump condenser replaces the resistance element in a joule-heated dryer, it is straightforward to calculate the energy cost for electrical resistance dryers. It can be seen that the energy cost for a comparable I^2R system is,

$$2.31\ (\text{COP}_s) \times \frac{\text{716 Btu supplied}}{\text{lbm water removed}} \times \frac{\text{6.5 lbm water}}{\text{bu}} \times \frac{\text{1 kWh}}{\text{3413 Btu}}$$

$$\times \frac{3.0¢}{\text{kWh}} = 9.45\ \frac{¢}{\text{bu}}$$

Consider now an LPG combustor in place of the heat pump condenser. If gas is 35¢/gal, has a heating value of 95,000 Btu/gal, a combustion efficiency of 92%, and the fan supplies a measured 27% of the required heat input, then the energy cost for an LPG low-temperature dryer is,

$$2.31\ (\mathrm{COP}_s) \times \frac{716\ \text{Btu supplied}}{\text{lbm water removed}} \times \frac{6.5\ \text{lbm water}}{\text{bu}}$$

$$\times \left[\frac{1\ \text{kWh}}{3413\ \text{Btu}} \times \frac{3.0¢}{\text{kWh}} \times .27 + \frac{1\ \text{gal LPG}}{.92 \times 95000\ \text{Btu}} \times \frac{35¢}{\text{gal}}\right.$$

$$\left.\times\ (1 - 0.27)\right] = 5.69\ \frac{¢}{\text{bu}}$$

Clearly the heat pump dryer is more economical to operate than the I^2R unit and even superior to the LPG system which consumes artificially inexpensive fuel. These results are shown graphically in Figures 5-3, 5-4, and 5-5. Of these figures it is especially interesting to note in Figure 5-5 the large difference in component energy consumption. The heat pump consumption is many times smaller than either the I^2R element or the LPG combustor and somewhat smaller than the fan. It is also evident from the figure that the heat pump as a heat source results in an absolute 32% reduction in energy consumption, assuming a 33% electrical generation and transmission efficiency. In addition, the heat pump drying system reduces the distribution load by a factor of almost 2.5 as compared to an I^2R system. This is of considerable importance to the Rural Electrification Cooperatives.

As with many new systems, the heat pump grain dryer did manifest some problems. It was initially constructed with a control logic which was intended to provide automatic operation. If the ambient temperature exceeded 70°F, a thermostat stopped heat pump operation until the temperature cooled below 65°F, where operation would automatically resume. This mode was designed to inhibit spoilage which rapidly occurs above 60-70°F. If the evaporator inlet air temperature (dryer volume effluent) fell below 47°F, then heat pump operation was automatically terminated to preclude frost formation. Normal operation resumed when the air temperature warmed to about 49°F (the dead-band was necessary to prevent harmful compressor cycling).

Curve 694845A

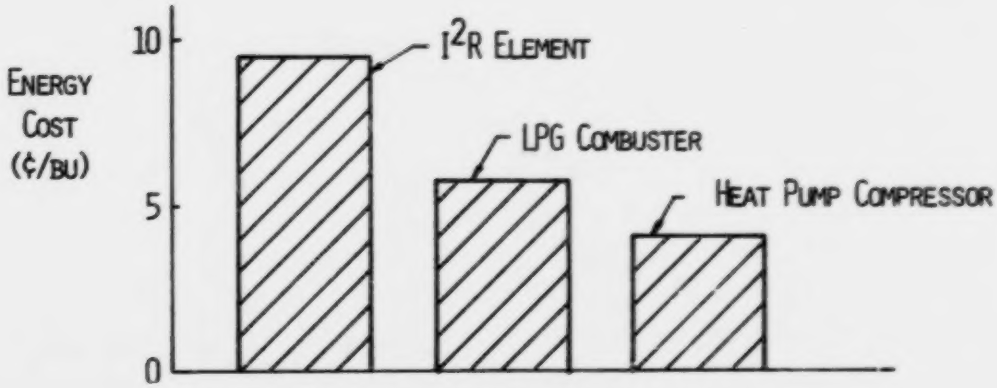


FIGURE 5-3. SYSTEM ENERGY COSTS.

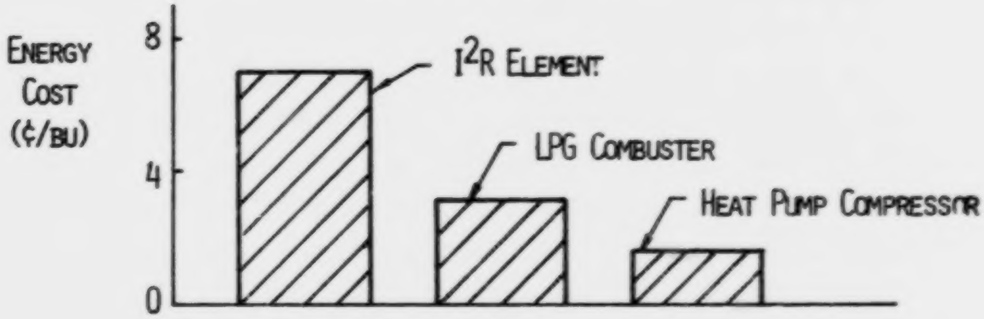


FIGURE 5-4. HEAT SOURCE ENERGY COSTS.

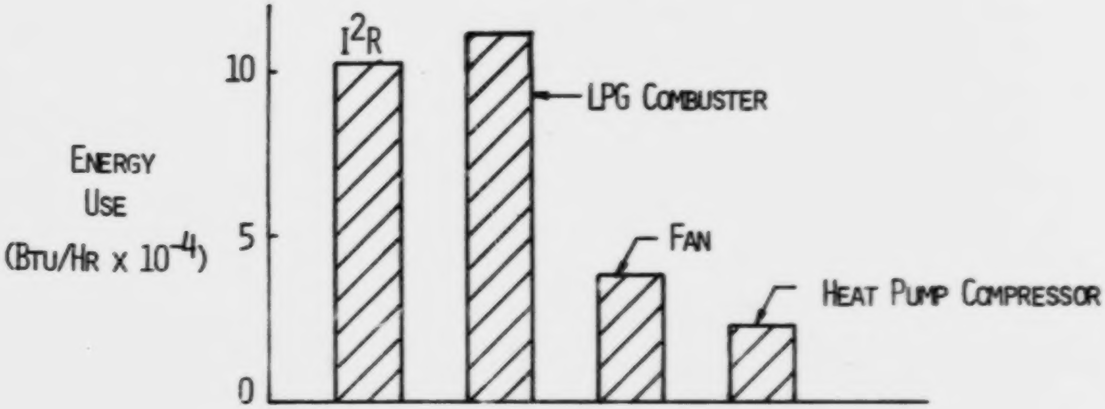
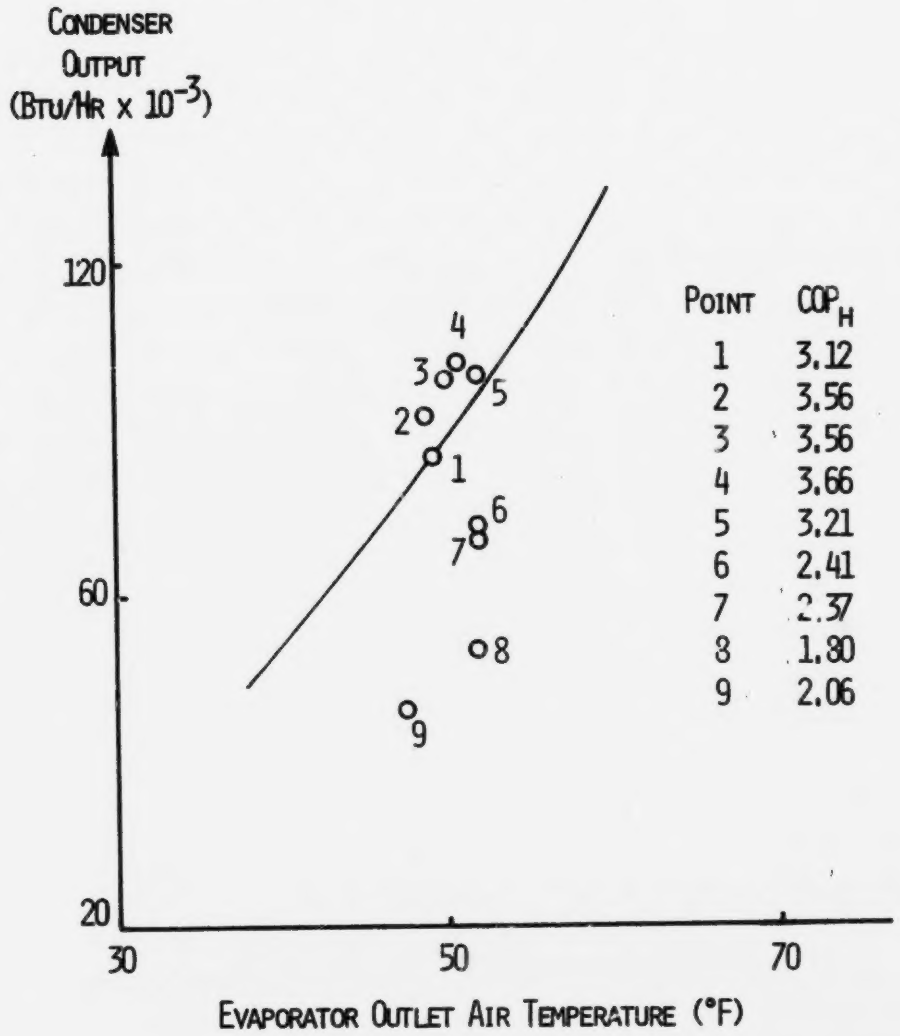


FIGURE 5-5. COMPONENT ENERGY CONSUMPTION.

The 1976 drying tests began with this automatic sequencing of operation. However, the grain was very cold when loaded and the weather continued to be unseasonably cool. Several days passed with no heat pump drying because of the abnormally low ambient temperatures. Even though the ambient temperature warmed to well above the required 45°F, which results in frost-free operation if the corn mass is sufficiently warm, it was not possible to operate the heat pump because of the large time lag between ambient and the air temperature exiting the grain mass. When it was evident that the system was not automatically coping with the unseasonal weather, system management reverted to manual control. The heat pump was manually started, overriding the automatic controls, and left running until the low pressure refrigerant control terminated operation because of excessive coil frost formation. Natural defrost then occurred, and the unit was restarted after complete defrost. Although the grain was successfully dried in this manner, it is an unacceptable method of operation because of the excessive amount of management required. In addition, the farmer does not want his drying operation impeded by adverse weather conditions any more than is already the case.

Figure 5-6 illustrates the deleterious effects of frost on heat pump capacity and performance, an effect well known in residential heat pumps. As seen in the figure, the data points are numbered from one to nine. Points 1 through 5 were recorded during frost-free operation and correspond closely to the expected performance shown by the solid curve. Points 6 through 9 were measured during coil frost formation. Clearly the heat pump capacity suffers, as does the heat source coefficient of performance, listed to the right of the curve. The coefficient of performance declines because of the increased temperature difference between the high-temperature sink and low-temperature heat source, whereas the capacity degradation is a result of the increased pressure ratio and reduced refrigerant density at the compressor inlet.

Curve 694846A



POINT	COP_H
1	3.12
2	3.56
3	3.56
4	3.66
5	3.21
6	2.41
7	2.37
8	1.80
9	2.06

FIGURE 5-6. EFFECT OF FROST ON HEAT PUMP PERFORMANCE.

Although frost formation must be tolerated in some heat pump applications where low-temperature operation is necessary, as is the case with residential heat pumps, this is not the case in low-temperature corn drying. Figure 5-7 illustrates the reason. Shown is a bar graph of the number of hours that the ambient temperature was in a given 5°F temperature range during the October 1 to November 15 drying season in 1977. This year can be considered as typical in that its average temperature is the same (49.5°F) as the 1974 through 1977 four-season average, and only during 1976 did the single-season average fall below 49°F, down to 42.9°F. Clearly the 45-60°F range exhibits the most activity, and, as a matter of fact, over 50% of the season exhibited an ambient temperature greater than 45°F. Since most of the drying occurs near 50°F, the heat pump must operate efficiently and at capacity at this temperature. For typical low-temperature drying conditions, this ambient temperature results in a grain effluent (evaporator inlet air) temperature of about 50°F. This is sufficiently distant from 32°F to allow frost-free operation of a reasonably sized evaporator. Thus, corn drying is one application where the design point excludes the debilitating effects of frost formation.

Although it has been shown that the design point can exclude the necessity of dealing with frost, there remains the problem of off-design operation which in some instances can account for 40% of the total operating hours. Several alternative methods for dealing with icing have been studied. These include timed operation in which the heat pump operation is terminated for natural or electric defrost; a control logic which prevents heat pump operation during icing conditions, yet maintains the grain column at the most favorable temperature for resumption of heat pumping; and the closed-air-loop system in which a fraction of the evaporator exhaust air is recirculated. The latter scheme is the most attractive in that it exhibits the best energy utilization and provides for continuous drying.

Curve 694847A

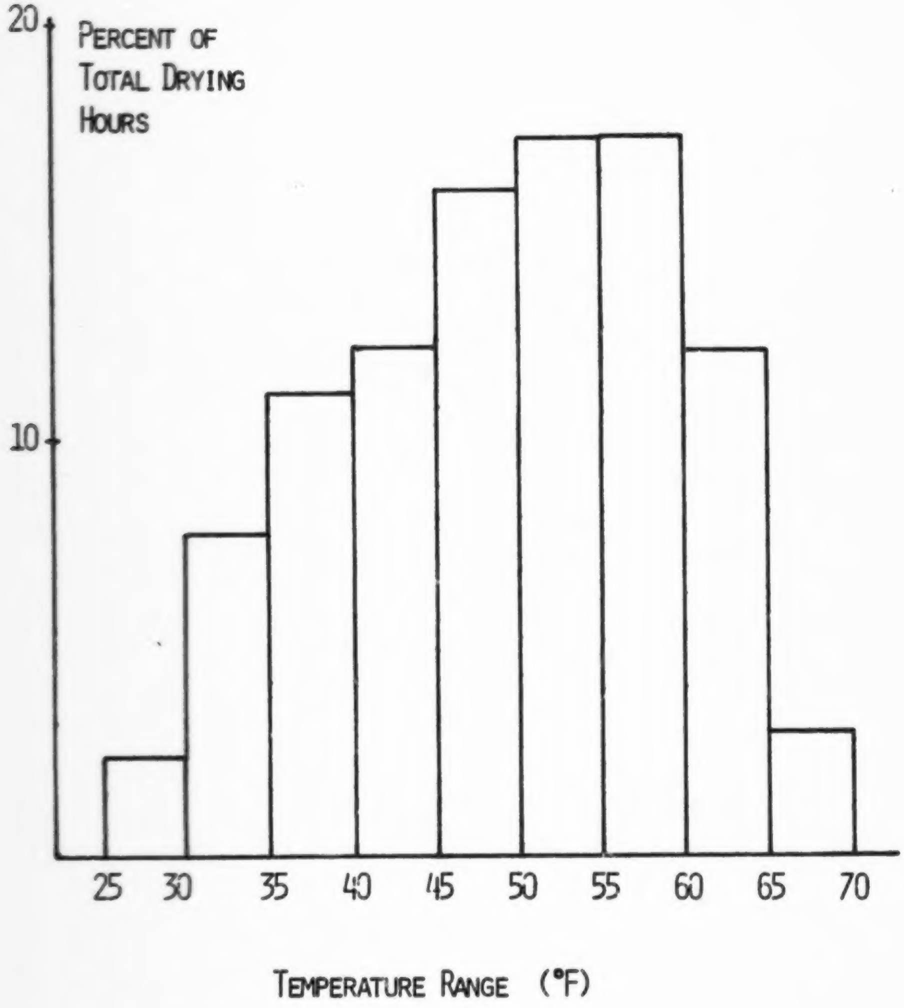


FIGURE 5-7. AMBIENT TEMPERATURE HISTOGRAM OF 1977 DRYING SEASON.

Consider Figure 5-8 where a schematic of the closed-air-loop system with its corresponding air states on a psychrometric chart is shown. Point A corresponds to the condenser inlet air state; B, the outlet air state; C, the grain column exhaust; and D, the evaporator outlet air state. In addition, state D is the minimum evaporator exhaust temperature before frost forms. The dashed line represents the locus of states on which A, the condenser air inlet state, must lie to maintain state D at the minimum design value (i.e., no frost). If state A moves to the right of the dashed line, then the evaporator exhaust air state, point D, will travel up the saturation curve and away from the minimum design temperature. However, if state A drifts to the left of the dashed curve without any action taken by the damper, then D will move below the minimum design value and frost will begin to form on the evaporator heat transfer surfaces. Consider now Figure 5-9 which illustrates the corrective action taken by the exhaust gas recirculation system. Since the locus of condenser air inlet states is approximately an isoenthalpic process, any deviation to the left of the curve can be detected by a wet bulb measurement. Corrective action can then be taken which would result in the clockwise rotation of the damper, mixing a portion of the evaporator exhaust airstream with ambient air. The resulting mixture (state A) would again lie on the locus of minimum condenser inlet air states as shown on the psychrometric chart, preventing frost formation. The advantages to this type of system are clearly evident. The heat pump can always operate without the deleterious effects of accumulating frost on the evaporator surfaces, no down time is incurred as a result of defrost or to avoid frost formation, no auxiliary energy is required for frost removal, and the dryer is always providing an air state to the grain column (state B) which results in a good drying rate.

A second generation low-temperature heat pump grain dryer is currently being developed under the sponsorship of the Department of Energy and will benefit from the previously-developed design tools as well as the three year's operational experience derived from the prototype grain

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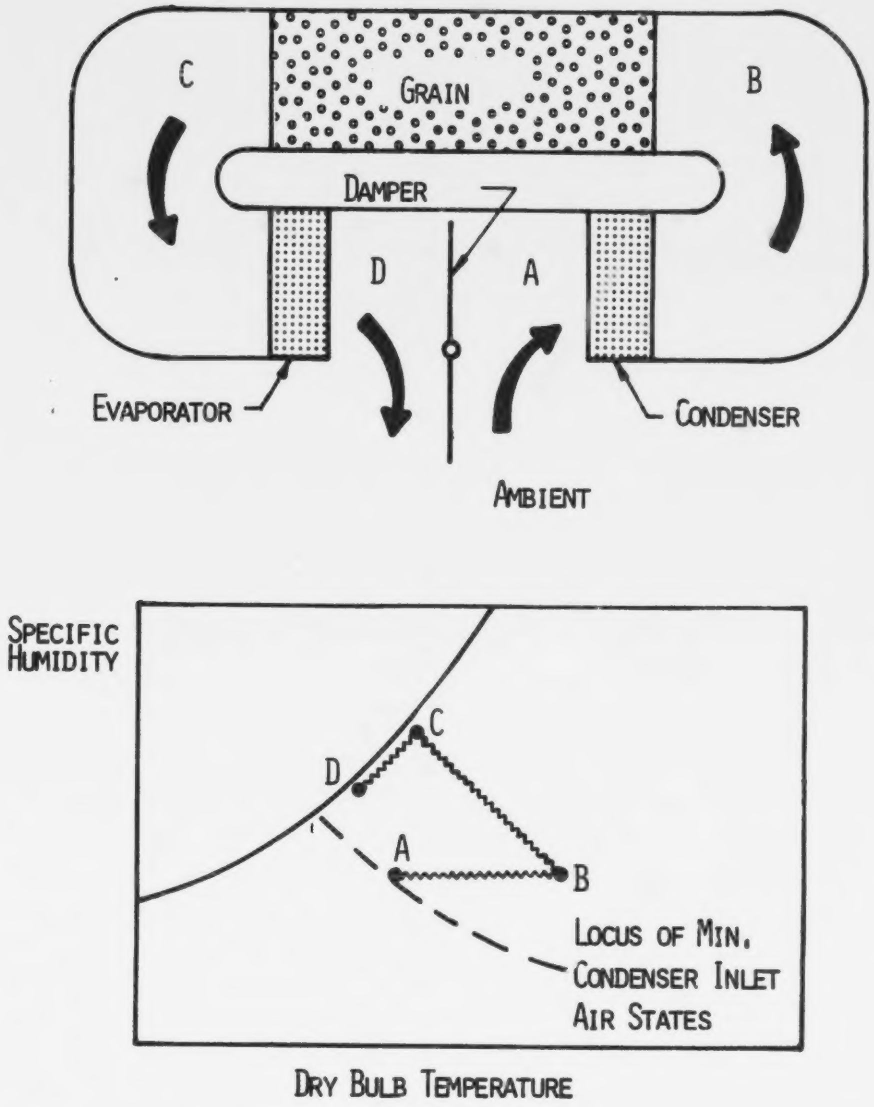
C
GRAIN
B
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EVAPORATOR
CONDENSER
AMBIENT
SPECIFIC HUMIDITY
C
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LOCUS OF MIN. CONDENSER INLET AIR STATES
DRY BULB TEMPERATURE

FIGURE 5-8. MINIMUM AIR STATE POINTS FOR NO FROST IN A CLOSED-AIR-LOOP SYSTEM.

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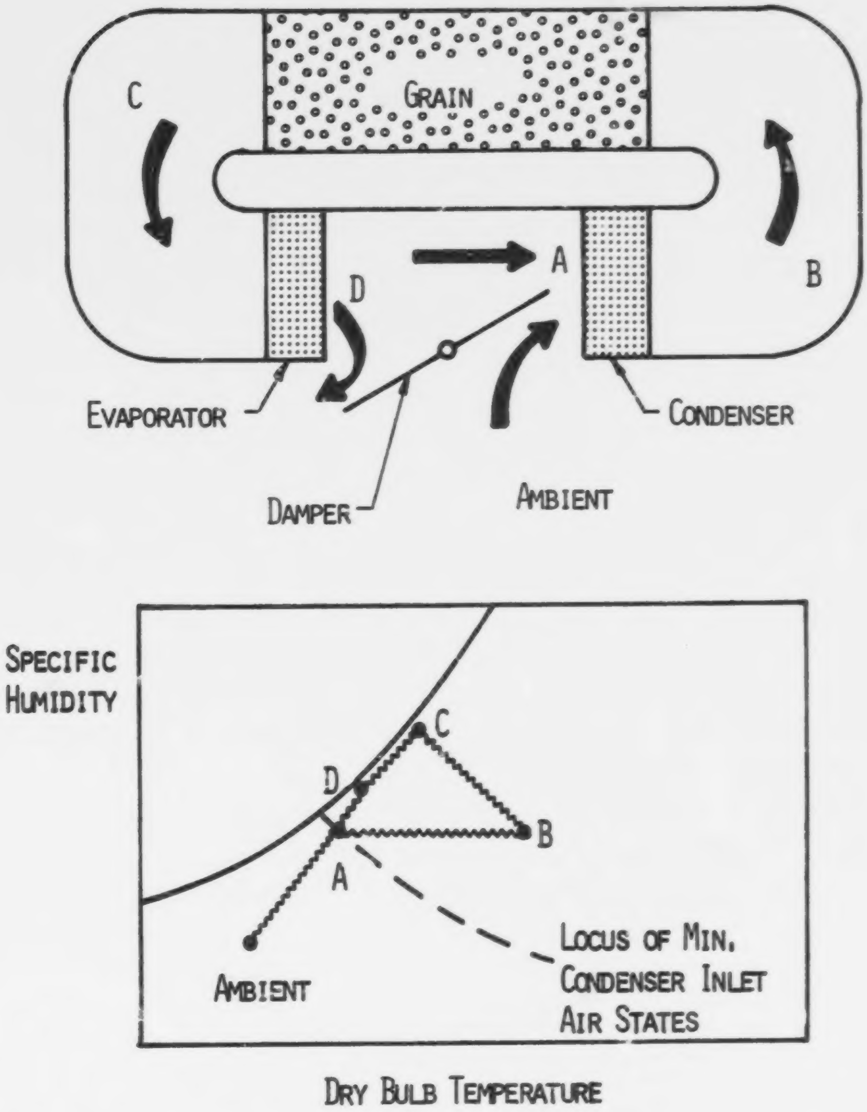
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LOCUS OF MIN. CONDENSER INLET AIR STATES
DRY BULB TEMPERATURE

FIGURE 5-9. PREVENTION ACTION FOR NO FROST IN A CLOSED-AIR-LOOP SYSTEM.

dryer. A major innovation is the incorporation of exhaust gas recirculation for below-design ambient temperature operation. In addition, the heat pump will be of unitary design. This will enhance the reliability of the heat pump grain dryer to the exemplary level exhibited by refrigerators and window air conditioners. It will also eliminate the relatively long refrigerant lines, thus improving capacity and performance. Better utilization of the performance characteristics of the centrifugal fan, used in deep bed drying, will allow for a more compact and economical heat exchanger design.

Another important result of the DOE-sponsored study will be a detailed evaluation of the economic viability of low-temperature heat pump grain dryers. During this phase, several competitive heat pump dryer designs, conventional I^2R, solar, and propane dryers will be compared. The heat pump design code will be used to size the heat pump units. The experience of Westinghouse in estimating the marketing price will be utilized in determining the farmer's initial cost for heat pump dryers. Manufacturers' data for conventional dryer types will determine their first cost, whereas the solar grain drying research currently being performed by Purdue University will be used in estimating the capital cost of potential solar dryers. First cost amortization will be factored into the economic evaluation with several projected energy cost scenarios, as well as maintenance and insurance costs, and income tax credits. The results of this program will become available in early 1979.

SECTION 6.0

SUMMARY

The uncertain supply of the convenience fossil fuels consumed in current grain dryers provided the initial impetus to examine alternative heat sources for drying. However, recent developments in the supply of energy require energy conservation in the absolute sense, i.e., reductions in the consumption of the energy-producing fuel. This, along with the risk-avertive nature of grain production, necessitates an efficient alternative which utilizes a reliable energy source. Low-temperature I^2R dryers have satisfied the criterion of a reliable energy source, and, to an extent, provided an efficient alternative, relative to high-temperature, large capacity units. Solar-heated, low-temperature systems are efficient but suffer from the uncertain availability of insolation. The low-temperature heat pump dryer, on the other hand, excels where the other methods suffer. It utilizes electrical energy, thus diversifying the energy source and utilizing fuels which, if consumed directly, would be unacceptable for grain drying. It is energy efficient in that it consumes one-fourth to one-sixth less electrical power as compared to an I^2R element. In addition, the heat pump as an alternative has a high probability of acceptance by agriculture because it utilizes a stable, reliable energy source.

Initial work with a prototype low-temperature heat pump dryer has demonstrated a COP_h of approximately 4.5; a drying efficiency of 716-760 Btu/lbm of moisture removed, as compared to 1500 Btu/lbm for I^2R units and 2500-3000 Btu/lbm for high-temperature dryers; and an energy cost of 4.09¢/bu, as compared to 5.69¢/bu for LPG and 9.45¢/bu for I^2R systems. This work has also shown the need for system modifications to preclude the unacceptable effects of frost formation on the heat pump heat transfer surfaces during unseasonably cool weather.

Under Department of Energy sponsorship, the Westinghouse Electric Corporation and the Purdue University Schools of Agriculture and Agricultural Economics will address the problems encountered with the prototype heat pump dryer and construct a demonstration unit to determine the viability of the heat pump grain dryer as an alternative to current systems. A thorough economic study will be conducted to determine the cost of operation and ownership of a heat pump dryer as compared to conventional systems and other potential alternative drying schemes.

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MANAGEMENT TECHNIQUES IN WASTE HEAT UTILIZATION

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SPONSORED BY ENGINEERING FOUNDATION
NEW ENGLAND COLLEGE
HENNIKER, NEW HAMPSHIRE
AUGUST 17, 1978

It is my pleasure to participate in this conference on waste heat utilization, a subject which has received and will continue to receive a great deal of attention as energy supplies dwindle and costs continue to rise. I'd like to spend the next twenty minutes or so describing Allied Chemical's corporate-wide energy conservation program and while doing this, I'll emphasize management techniques which have been successfully used in improving our energy performance and in particular the utilization of waste heat.

First let me give you some background on Allied Chemical. Allied is a diverse company producing a wide variety of products ranging from basic industrial chemicals and specialty chemicals, to seat belts, synthetic fibers, coke and agricultural chemicals. We also produce oil, coal, natural gas, and nuclear fuels, and have developed and built a plant for processing spent nuclear fuels. Allied operates in over 120 locations which vary in size employing anywhere from a few people up to about 2,500.

In 1973, our energy bill was a little over $70 million, last year, with a reduced level of consumption of fuels (on a Btu/lb basis), this bill has escalated to more than $300 million. Since 1972, we have been engaged in a vigorous effort to improve the energy efficiency of our operations. As a result of these efforts over the last five years, we have improved our efficiency by about 14% as compared to 1972 with an estimated savings of about $40 million in 1978.

Since the beginning of our efforts we have developed and instituted what we believe to be the essential management techniques required to realize substantial energy savings.

These are:

- Commitment
- Organization
- Identification
- Communications
- Implementation
- Accountability

These essentials, are applicable not only to our large and diverse operations but to a wide variety of smaller companies, institutions, and businesses as well who have an interest in improving their energy efficiency with the resultant cost saving benefits.

COMMITMENT

The first essential to any effective energy conservation program is commitment. By commitment I mean that at the upper management levels within a company there must be a conscious and visible dedication to energy conservation. Since companies differ widely, the form of this top management commitment will vary. At our company, this commitment was appropriately expressed by the issuance of a policy statement by the Chairman of the Board (Exhibit A).

This statement of policy clearly focuses the attention of all levels of our management on the emphasis and importance that our top executives place on conservation.

But commitment goes beyond just an expression of policy. Company management must become truly dedicated to energy conservation, giving it the same emphasis it places on other profit-oriented company activities.

ORGANIZATION

Once the initial management commitment is made, the next essential element of our program was the development of appropriate organization concepts and procedures. To expedite programs for conservation at Allied Chemical a corporate department was created for energy resources management with one emphasis being the initiation of energy conservation programs.

Like many large companies, we are made up of a number of operating divisions. To permit effective coordination of company programs, senior management representatives were selected from each division to form the Corporate Energy Council. To promote better communication on energy matters, each operating division developed organizational relationships between operating locations and the Division Energy Council representative. Each plant has a reporting relationship to it's respective Division Energy Council representative through a Plant Energy Committee. The Plant Committee is chaired by the plant energy manager. It should be noted that all of the important plant functions are represented on the committee to permit effective implementation of the company's programs.

IDENTIFICATION

In addition to commitment and organization, another essential component of effective energy conservation is identification or determination of opportunities for energy savings. This phase of any conservation program is important because it establishes the magnitude of possible savings and this, of course, provides the real incentive for the plant program. It should be realized that energy saving measures can be identified by many means. These include formalized energy audits, equipment reviews, and technology and operating assessments involving all levels of personnel.

A good starting point for us in our Identification Program was the energy audit. An audit allows relatively quick assessment of the magnitude of savings achievable and thus helps to establish priorities. Our corporate approach to audits was to select 32 of our over 120 operating locations for initial auditing. These locations were selected because they account for more than 90% of Allied's total energy bill.

Plant energy audits were conducted through a cooperative effort between a task force of engineers from corporate offices and plant engineers. Each audit started with the completion of a questionnaire and a plant visit which took from two to as many as seven working days. Upon completion of the plant visit, a comprehensive report was prepared. Each report includes an item by item identification of each conservation opportunity, the savings potential in both Btu's and dollars, as well as an order or magnitude cost estimate and simplified rate of return on the project if capital funding is required. Through our audit program many areas to improve waste heat utilization were identified.

From the 32-plant energy audits conducted during late 1973 and throughout 1974, a total of over $25 million per year of potential energy savings in 1974 dollars were identified.

One major side benefit of our audit program was the identification of many repetitive areas condusive to energy savings. These areas, many of which are waste heat oriented, had such wide-spread application within Allied that specific guidelines were prepared to facilitate rapid implementation at each of our plants. These guidelines were compiled into an Energy Conservation Guidebook for distribution to the energy manager at each plant. The Guidebook serves as a reference source of techniques for quickly evaluating energy conservation opportunities. For ease of use at the operating level, each section is arranged to:

1. Identify where opportunities exist.

2. Describe how to test for relative efficiency of the unit operations.

3. List modifications and/or additions which will result in maximum economy.

4. Provide guidelines for maintaining maximum efficiency on a day-to-day basis.

Let me present two examples of the use of the Guidebook pertinent to waste heat utilization.

A prime factor affecting either boiler or process furnace efficiency and ultimately fuel consumption is the temperature of combustion gases rejected to the stack. Increased operating efficiency with a corresponding reduction in fuel input can be achieved by rejecting stack gases at the lowest practical level consistent with basic design principles.

The Guide first lists the major equipment in the plant which can benefit from heat recovery (such as steam boilers, process fired heaters, and other heat transfer furnaces) and gives overall guidance for potential application, namely a flue gas temperature above approximately 250°F. In addition the major types of heat recovery equipment are identified and briefly described; in this case the application of air preheater, economizers or similar exchangers. Planning quality guidelines are then presented to allow the plant engineer to quickly determine final sink temperature and to determine comparative economic benefits to be derived by the installation of heat recovery equipment. After determining a few key parameters, procedure are given which allow calculation of the approximate fuel savings.

In the example shown in Exhibit B the unit heat output of 150 million Btu per hour is known, exit gas temperature of 670°F is measured, and a desired gas temperature of 330°F has been determined from a table in the Guide. Then using the nomograph shown in Exhibit C, savings of $112,000 per year for fuel oil at 75 cents per million Btu are rapidly calculated. This data combined with other considerations for equipment application (such as space availability, type of fan, and maintenance considerations) allow determination of the feasibility to proceed with more detailed analysis.

The next example follows essentially the same procedure for minimizing water being blown down from a unit. Areas of conservation to be considered include:

- Minimize blowdown by automatic or manual adjustment
- Recovering more clean condensate
- Increase allowable solids level
- Modify water treatment
- And again, heat recovery.

Illustrations, such as the one in Exhibit D, emphasize the importance of maintaining the optimum solids level. As can be seen operation at a lower drum concentration than necessary (3000 ppm instead of 6000 ppm) results in a much higher blowdown flow rate, 4.5% versus 9.5% and consequently higher heat loss. The nomograph in Exhibit E then allows quick determination of annual losses through relatively simple changes in operating procedures. In the example shown blowdown was reduced from 12% to 10% on a 100,000 lb/hr, 150 psig boiler with fuel costs at $1.00/MM Btu. Annual savings are found to be $800/yr. Standard types of heat recovery systems such as the two-stage heat recovery system in Exhibit F are also presented in the Guide for further consideration by the plant.

Another essential component of our identification program has been technology assessment. The production of chemicals is one of the most energy-intensive industries. With the rising cost of energy, the question arises whether new or modified technology can be developed which is measurably more energy efficient and, therefore, produces chemical products at lower cost. Recently a program of technology assessment of Allied's processes has been instituted to permit judgements to be made in this area.

The assessment program provides the opportunity to draw upon the best technical talent in our company to critically review our chemical operations from the standpoint of energy efficiency. The five steps involved in each process assessment include:

1. Calculate Direct Process Energy (DPE).
2. Assemble Information Package.
3. Brainstorming Session.
4. Technical and Economic Evaluations.
5. Final Report.

The program began with calculation of the direct process energy or DPE requirements for each of 24 main processes used by the company. This information was compiled by our Engineering Technology Center who presented a report with recommendations to the Energy Council. The Council selected five processes for initial study, however an additional five processes have recently been added. Selection was made based upon consideration of total energy required for production, viability of product, high energy consumption on a Btu per lb basis, and likelihood of major energy reductions being possible.

Following the initial selection, packages of information were assembled for each product. These packages include; step-by-step energy consumption profiles for the process, type and quantity of each fuel used, flow diagrams for each process stage, existing energy-conservation programs at the plant, and published information on rival routes. This information was put into a briefing book and normally took four to six weeks to compile and would run 40 to 100 pages depending on process complexity.

The briefing book set the stage for the major brainstorming session on each process. Typically 10 to 15 participants were invited to each session. These people came from all locations, and management levels of the firm. For instance, the TDI group included several process-development engineers, an energy-conservation engineer, both the director and associate director of the Engineering Technology Center as well as its manager of process engineering, a divisional research chemist, and also the plant superintendent and technical supervisor. One or more people from each plant using the given process were asked to take part in each of the discussions. The objective has been to assemble a panel with a broad range of background and experience to help assure a wide spectrum of inputs on the various points brought up.

Brainstorming sessions typically lasted from one to two days and encouraged each person present to participate. Each area of the information package was thoroughly reviewed and a lot of re-examining of the way things were being done occurred. From the brainstorming session many ideas were generated.

After the brainstorming, the program's fulltime staff evaluated the technical aspects and preliminary economics of all the suggestions for a given process, and issued a draft report of their findings. Typically, this draft required six to ten weeks to prepare.

All the reports place suggestions into three categories: Class I items are technically feasible, and require only routine engineering effort and more-detailed costing with little or no process change; Class II items are technically feasible, but require some process change and modest development or further engineering and economic analysis; Class III items require fundamental process changes, substantial engineering and in-depth economic study -- also potentially feasible if extensive basic research and development were to be carried out. For certain processes, each category could be subdivided into application to existing units or to new or expanded units only.

Illustrating this grading for a phenol process indicated that a waste heat exchange system between an effluent stream and a dehydrator feed stream rated as Class I (Exhibit G) while Class II items (Exhibit H) included a two-stage preflash system, column modification, and steam jet modifications since modest process modifications are required and more detailed analysis. Items shown in Class III (Exhibit I) required substantial modifications or more development.

The Class I items obviously represent savings potentially achievable in the near term. This category also identified a large number of waste heat applications. The five processes evaluated so far promise from 5 to 10% cuts in energy consumption if all pan out. Some of the suggestions already are being implemented. In the future, Class II ideas may possibly provide 3-14% additional savings. Class III items would add up to 25% to these savings.

IMPLEMENTATION

After energy saving opportunities have been identified, effective implementation becomes essential in order to realize maximum bottom line savings. Since the energy savings opportunities vary widely in character and type, the implementation phase requires a number of individual and specific actions. In many cases improvement in energy efficiency can be accomplished through training and such was our experience primarily in our powerhouses and other high energy consuming operations where precise control is essential. Major savings were realized through careful and conscientious review and upgrading of our operating procedures and retraining our supervision and operators to meet the more demanding energy economics.

In a number of cases, implementation of energy savings can also be achieved through improved maintenance and this was particularly evident with regard to steam leaks, steam traps and other repair-type of activity. Sections of the previously mentioned Energy Guidebook were devoted to steam leak and steam trap maintenance with particular emphasis on developing an understanding among our operating people on the significant cost impact of energy losses resulting from poor maintenance and operating practices. To expedite this effort in our plants, actions such as identification of maintenance people to work full time on upgrading of energy systems was taken. Major savings have been realized with programs of concentrated maintenance efforts on the integrity of our steam distribution systems and facilities.

In addition to various training and maintenance improvements, it was essential that capital programs to install more energy efficient equipment and new equipment for such things as heat recovery and improved control be undertaken. With rising fuel costs, these capital investments can, in most cases, be readily justified on the basis of the savings to be realized with their installation.

COMMUNICATIONS

As our overall energy conservation program became more sophisticated, it became apparent that maximum savings could only be maintained through the efforts of all employees. In 1976 an intensive communications and awareness program was formalized with the objective of making each of our 33,000 employees aware of the importance of saving energy at home and on the job. An outside consultant specializing in employee communications was retained to determine the general level of employee awareness to the energy problem and to create a program to stimulate employee cooperation in the conservation of energy. Formal in-depth research was completed which revealed that, like most other Americans, the Allied Chemical employees interviewed had a relatively low awareness of the magnitude of the energy problem and little understanding of how they would be affected.

Based on this knowledge, our "PULSE" program was developed. "PULSE" is an acronym for Please Use Less, Save Energy. It is an incentive-based communications program designed to stimulate individual awareness of how energy shortages and sharply rising energy costs will affect company operations and each individual.

Prior to the formal initiation of the program, a pre-program "teaser" campaign was initiated calling attention to the "PULSE" logo without identifying its meaning. Ads were run in the company newspaper, posters appeared on plant bulletin boards, and a message was distributed with employee paychecks.

To introduce the program to plant management, a day-long kickoff meeting was held at our corporate headquarters. This meeting included talks on the program's importance by our Chairman of the Board and our President. Descriptive materials of the "PULSE" program were provided, including a plant manager's guide. With managers aware of the elements of the program and their role in it, individual plant kickoff programs were then scheduled in Allied Chemical locations. Each employee received a booklet outlining his critical role in the success of the "PULSE" program.

Central to the "PULSE" program is the stimulation of employees to submit energy-saving ideas to their plant management for review and implementation. Energy-saving idea forms were provided for employees to submit suggestions and at the plant management discretion, outstanding suggestions by employees for saving energy are rewarded with a certificate of commendation as well as an individual plant-provided award. The program, during the first year, generated the submittal of over 5,000 employee suggestions of which more than 2,000 were accepted by the plant energy review committees as technically feasible.

Throughout the program, numerous communications materials constantly remind employees of the need to save energy. These include posters, decals, floor stencils, special bulletins, pay envelope stuffers, banners, flags, and extensive use of in-house publications. As a matter of fact, just about every means available is used to promote the "PULSE" energy conservation program.

This high level of employee response created considerable interest in the program by many of the communities in which we operate. As a result of this interest, the program has recently been extended beyond our plant fencelines to include the communities. This program extension encourages our plant management groups to work with schools and civic organizations in their local communities on programs to promote energy conservation. Films and other visuals are provided permitting our management representatives to speak at local functions and at local schools on the national energy problem and on ways in which individuals can make a meaningful contribution to the solution of these problems. This effort has produced results beyond initial expectation in terms of invitations to participate in the community activities related to energy.

Involvement of the individual is essential to the success of any communications program. "PULSE" is intended to do just that, to bring as many employees as possible together in a common effort to conserve energy.

ACCOUNTABILITY

Once having set forth on a program of improving energy use efficiency, it is essential that some accountability and measurement of progress on energy conservation be developed and instituted. We have adopted a quarterly report which provides a rolling assessment of the energy consumption per unit of output for an annual period. In addition, this report provides information on specific energy form and usage as well as the status of energy projects underway in the individual plants. From this data we prepare a semi-annual report for our trade association, the Manufacturing Chemists Association, who in turn accumulate the data on energy use per unit of output for chemical companies in the U.S.

In conclusion, these are the essentials that we feel are critical to the success of an energy conservation program. They are not particularly unique and, in most cases, simply represent logical management practices. Not all of them, I am sure, apply to each and every company and perhaps there are some essentials which we have overlooked, but in our experience, application of these essentials has produced significant cost savings and measurable improvement in energy efficiency.

CORPORATE POLICY

IT IS THE POLICY OF ALLIED CHEMICAL CORPORATION:

1. TO CONSERVE AND EFFICIENTLY USE THE NATION'S NATURAL RESOURCES.
2. TO OPERATE ITS PLANTS TO AVOID ENERGY WASTE AND TO MAKE THE MOST EFFICIENT USE OF THE FUELS AND FEEDSTOCKS WHICH CONSTITUTE ITS MAJOR ENERGY RESOURCES.
3. TO RECOGNIZE ITS OBLIGATION TO ITS SHAREHOLDERS AND TO ITS CUSTOMERS TO MAINTAIN THE PROFITABILITY AND CONTINUITY OF ITS PLANTS OPERATIONS. ACCORDINGLY, ALLIED WILL EXAMINE ALL ENERGY SOURCES IN THE INTEREST OF HAVING ALTERNATIVES AVAILABLE IN THE EVENT USUAL SOURCES ARE NOT AVAILABLE.
4. TO TAKE THE NECESSARY STEPS TO ENSURE THAT ALL OF ITS EMPLOYEES ARE COGNIZANT OF AND RESPONSIVE TO THE FOREGOING POLICY.

JOHN T. CONNOR
CHAIRMAN OF THE BOARD
APRIL 9, 1973

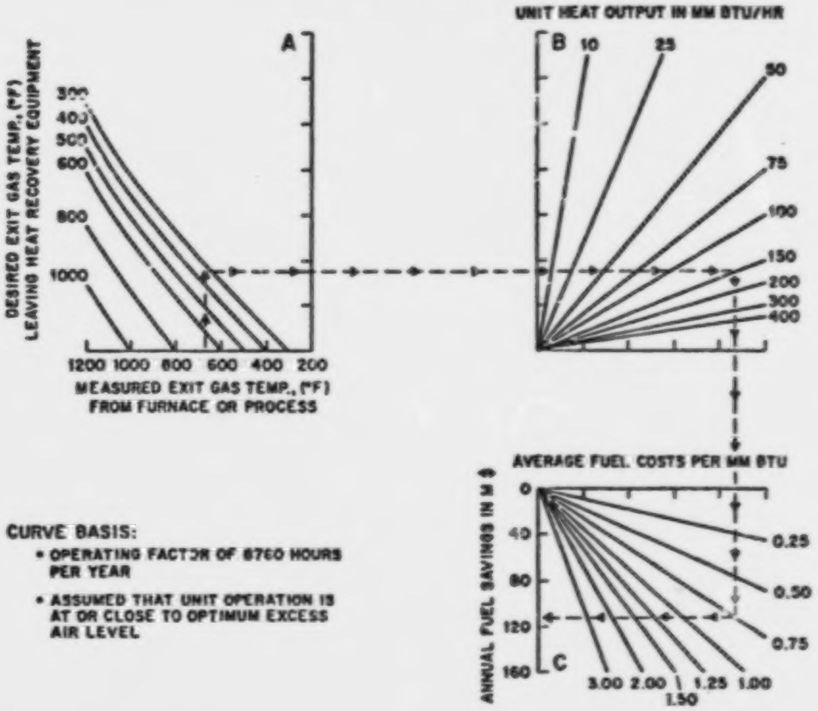
Exhibit A

REDUCED STACK TEMPERATURE – EXAMPLE

Unit Heat Output:	150MM BTU/HR
Measured Exit Gas Temp:	670° F
Desired Gas Temp:	330° F
Firing:	No. 2 Fuel Oil
Cost:	$0.75/MM BTU

Potential Annual Fuel
Savings = $112,000

NO. 2 FUEL OIL
ANNUAL SAVINGS FROM REDUCED STACK TEMPERATURE



CURVE BASIS:
- OPERATING FACTOR OF 8760 HOURS PER YEAR
- ASSUMED THAT UNIT OPERATION IS AT OR CLOSE TO OPTIMUM EXCESS AIR LEVEL

Exhibit D

REQUIRED PERCENT BLOWDOWN

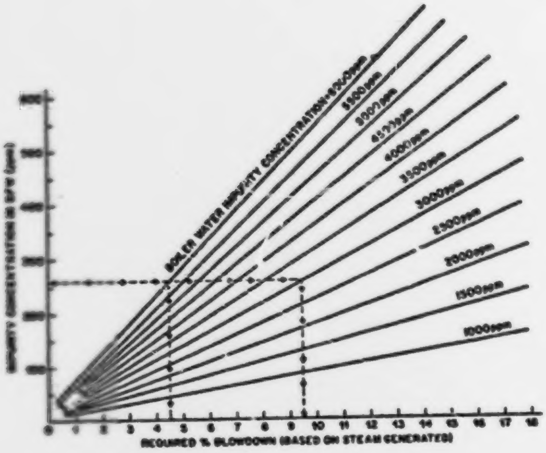


Exhibit E

HOURLY COST OF BLOWDOWN

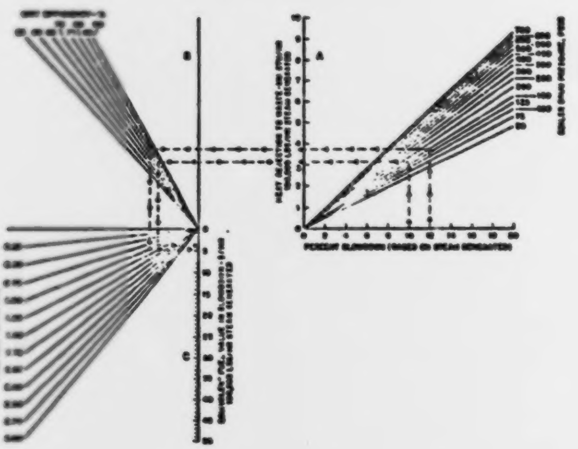


Exhibit F

TYPICAL TWO-STAGE BLOWDOWN HEAT RECOVERY SYSTEM

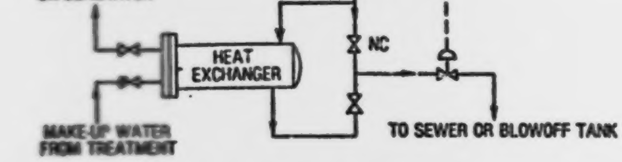

BOILER STEAM DRUMS
BLOWDOWN LINES
FLASH STEAM
TO PROCESS OR DEAERATOR
FLASH TANK
LEVEL CONTROLLER
LC
MAKE-UP WATER
TO CONDENSATE TANK
OR DEAERATOR
HEAT EXCHANGER
NC
MAKE-UP WATER
FROM TREATMENT
TO SEWER OR BLOWOFF TANK

Exhibit G

CLASS I ITEMS

Item No.	Description	Annual Savings MMM Btu	Annual Savings M$	Capital M$	DCF Summary Payback Years	DCF Summary % Return
I-1 – Unit 1	Dehydrator Feed/ Effluent Exchange	6.18	14	50	6.2	12.0
I-1 – Unit 2	Same	7.42	17	50	5.4	16.0
I-1 – Unit 3	Same	14.67	34	58	3.6	29.5
I-2 – Unit 1	Generate Low Press. Steam From Process Column Condensers	36.20	84	315	6.6	10.5
I-2 – Unit 2	Same	50.10	116	370	5.8	14.0
I-2 – Unit 3	Same	116.40	269	500	3.8	27.0

Exhibit H

CLASS II ITEMS

Item No. & Description		Potential Annual Savings MMM Btu	Potential Annual Savings M $
II-1	Two-Stage Preflash	132.0	309
II-2	Refining Column Alternatives	93.2	218
II-3	Reduce Size of One Jet by 50% for All Parallel Installations	60.0	140

The following Class II ideas are applicable only to new plants.

		MMM Btu Per Year
IIa-1	Preheat by Feed/Exit Air Exchanger	141.9
IIa-2	Use Reactor Effluent Heat for Absorption Refrigeration or Recovery via Heat Pump	134.5

Exhibit I

CLASS III ITEMS

Item No.	Description	Potential Annual Savings MMM Btu	Potential Annual Savings M $
III-1	Possible Yield Improvement from Catalysts	Yield	2,400*
III-2	Operate Decomposers at 130°C to Generate 15 psig Steam	794	1,860
III-3	Recover Decomposer Heat by Flasing By-Product	50% of above	
III-4	Combine Crude Columns	109	225

The following items were not evaluated.

III-6	Increase O_2 Concentration in Reactor Air
III-8	Debottleneck Recovery System
III-9	Repack Column to Reduce Reflux

*Includes Energy and Yield Benefits

SELECTION OF WASTE HEAT RECOVERY EQUIPMENT
IN
PETROLEUM REFINING

by: Roy W. Miller
Coordinator, Energy Conservation
Amoco Oil Company
Chicago, Illinois

Conference on Waste Heat Utilization
New England College
Henniker, New Hampshire
13-18 August 1978

SELECTION OF WASTE HEAT RECOVERY EQUIPMENT IN PETROLEUM REFINING

INTRODUCTION

THE PROPER SELECTION OF WASTE HEAT RECOVERY EQUIPMENT CAN MAKE THE DIFFERENCE BETWEEN HAVING A PROFITABLE REFINERY ENERGY PROJECT AND ONE THAT GIVES ENERGY CONSERVATION A BLACK EYE. IN MANY ENERGY CONSERVATION AREAS, A NUMBER OF HEAT SAVING OPTIONS ARE AVAILABLE TO REFINERS BUT OFTEN NOT A GREAT DEAL OF THOUGHT IS GIVEN TO THE PROPER SELECTION OF THE EQUIPMENT. AS A RESULT, THE PROJECTED PAYBACK FROM THE CAPITAL INVESTMENT MAY NOT BE REALIZED. A NUMBER OF FACTORS BEYOND THE DESIGN CRITERIA MUST BE CONSIDERED TO MAXIMIZE THE LONG-TERM RETURN ON CAPITAL.

WASTE HEAT RECOVERY EQUIPMENT

WASTE HEAT CAN BE RECOVERED IN REFINING BY DIFFERENT TYPES OF FACILITIES, THESE ARE LISTED BELOW IN THE GENERAL ORDER OF PREFERENCE.

1. ADDITIONAL OR OPTIMIZED HEAT EXCHANGE SURFACE TO REDUCE HEAT LOSS TO COOLING WATER OR AIR.

2. ADDED CONVECTIVE SECTION IN FURNACES TO REDUCE STACK LOSSES TO ATMOSPHERE.

3. AIR PREHEATERS TO REDUCE FURNACE STACK LOSSES TO ATMOSPHERE.

4. Waste heat boilers to reduce furnace stack heat losses to atmosphere.

5. Low level energy recovery.

Also, waste heat generation can be reduced by controlling furnace excess air and air-in leakage. Power recovery is a separate category, and will not be evaluated herein.

Unit Energy Audit

When an energy audit is made of a petroleum refining unit, a set of criteria is assumed for best operation. These criteria probably have different values at different companies, but those listed below will not be too far from the norm. All reasonable best heat sinks should be investigated inside and outside the operating unit for available waste heat recovery.

1. Product streams to cooling water or air, NMT 250°F.

2. Furnace flue gas to atmosphere, NMT than 400°F.

3. Furnace excess air, NMT 20%.

4. Radiant heat surfaces, NMT 150°F.

New refining units should be designed to the above criteria, and old units should be retrofitted with waste heat recovery equipment to achieve these ends. As a general rule it is felt that for most refineries, energy reduction potentials are evenly split between the furnace stack flue gas and process streams going to cooling water or air.

Using the above criteria, waste heat recovery equipment is designed into new or existing facilities. This can conveniently be done on new units but retrofits become a problem, both money wise and space wise. New facilities can be designed for optimum energy recovery conditions (power recovery from FCU regenerator), whereas on old units, you take what you have.

Additional or Optimized Heat Exchange

Additional heat exchange surface is probably the best method for recovering waste heat on a petroleum unit. The major incentives include:

1. High return on investment.

2. Low maintenance cost, few or no moving parts.

THE RAPID RISE IN HEAT EXCHANGER COSTS AFTER THE ARAB EMBARGO HAS NOW LEVELED OFF OR IN SOME CASES DECREASED WHILE THE COST OF FUEL HAS BEEN CONSTANTLY ESCALATING.

ADDITIONAL HEAT EXCHANGE SURFACE IS NORMALLY ASSOCIATED WITH PREHEATED FEED STREAMS GOING TO A FURNACE, BUT IT ALSO SHOULD INCLUDE THE OPTIONS OF:

1. PREHEATING FUEL (ESPECIALLY FUEL GAS) BEING BURNED IN FURNACE (1# FUEL WITH 20# OF AIR).

2. PREHEATING COMBUSTION AIR GOING TO FURNACE WITH HOT OIL OR A SIMILAR STREAM WILL NOT GIVE THE HIGH COMBUSTION AIR TEMPERATURES AS FROM NORMAL AIR PREHEATER. PUT, THE INSTALLATION WILL BE CHEAPER AND SOMETIMES LESS RISKY TO OPERATE.

3. REBOILING FRACTIONATING TOWERS INTERNAL OR EXTERNAL TO THE UNIT.

4. BOILER FEED WATER PREHEAT OR OTHER REFINERY HEAT REQUIREMENTS.

OPTIMIZING EXCHANGERS SYSTEMS ARE NOW COMPUTER ORIENTED AND RAPID ADVANCES ARE BEING MADE IN THESE TECHNIQUES. THIS CAN BE VERY COMPLEX WHEN OPTIMIZING EXCHANGERS IS DONE IN CONJUNCTION WITH HEAT INTEGRATION BETWEEN UNITS.

ADDED CONVECTIVE AREA

ADDING FURNACE CONVECTIVE AREA IS ALSO A MOST REWARDING SCHEME, ESPECIALLY IF LOW LEVEL FEED PREHEAT IS NEEDED AND THE FURNACE CAN BE EASILY RETROFITTED FOR THE ADDITIONAL TUBES AND SOOT BLOWING EQUIPMENT.

ADDED CONVECTIVE SECTION CAN BE UTILIZED IN SEVERAL DIFFERENT MANNERS.

1. PREHEAT FEED GOING TO RADIANT SECTION.
2. GENERATE STEAM FOR UNIT OR REFINERY USE.
3. HEAT EXTRANEOUS STREAMS (REBOILING, AIR PREHEATING VIA OIL CIRCULATING STREAMS, BOILER FEED WATER PREHEAT, ETC.).

RETROFITTING CONVECTIVE SECTIONS IS OFTEN VERY DIFFICULT BOTH FROM A CONSTRUCTION STANDPOINT AND FROM A DIMINISHING RETURN EFFORT. ON THE PLUS SIDE CONVECTIVE SECTIONS HAVE NO MOVING PARTS AND ARE SOMEWHAT MAINTENANCE FREE. REBOILERS AND OIL TO AIR PREHEATERS MAY HAVE MOVING EQUIPMENT, AS WILL SOOT BLOWERS IF USED.

AIR PREHEATERS

AIR PREHEATERS ARE THE ULTIMATE WEAPON TO REDUCE FURNACE STACK TEMPERATURE. IN ADDITION, INSTALLATION OF AN AIR PREHEATER

WILL SOMETIMES GET MORE CAPACITY OUT OF A UNIT FURNACE THAN FORMERLY ACHIEVED. THE ADDED CAPACITY CAN ONLY BE GAINED IF OTHER FURNACE OPERATING PARAMETERS ARE NOT BEING EXCEEDED WHEN THE HOTTER FURNACE RADIANT SECTION IS ATTAINED. A NUMBER OF PITFALLS ARE ENCOUNTERED WITH AIR PREHEATERS.

1. EXCESSIVE CORROSION OFTEN OCCURS, ESPECIALLY WHEN SULFUR COMPOUND DEWPOINTS ARE APPROACHED.

2. SEAL LEAKAGE BETWEEN INCOMING AIR AND HOT FLUE GASES CAN REDUCE EFFECTIVENESS OF AIR PREHEATER.

3. DAMPER LEAKAGE IN DUCT WORK WHICH WILL RECIRCULATE FLUE GAS OR AIR STREAMS WILL CAUSE A LOSS IN EFFICIENCY.

4. AIR PREHEATERS ARE ALSO KNOWN TO CATCH FIRE BECAUSE OF:

    A. SULFUR DEPOSITS IN SYSTEMS.

    B. CARBON DEPOSITS IN SYSTEM.

    C. FURNACE FUEL GETTING INTO THE FLUE GAS (FUEL RICH SITUATION) AND THEN BURNING IN THE AIR PREHEATER. AS MORE AIR PREHEATERS ARE BEING INSTALLED, YOU WILL BE HEARING ABOUT MORE FIRES IN THESE SYSTEMS. IT WOULD BE WISE TO REVIEW YOUR SAFETY PROCEDURES NOW.

COMBINING ADDED CONVECTIVE AREA WITH AN AIR PREHEATER HAS BEEN PROPOSED FOR RETROFIT SITUATIONS. THE CONVECTIVE AREA WOULD BE AT GRADE EXTERNAL TO THE FURNACE. THIS COMBINATION WILL BE VERY EFFICIENT AND OVERCOME A MULTITUDE OF CONSTRUCTION AND OPERATING PROBLEMS.

WASTE HEAT BOILERS

WASTE HEAT BOILERS ARE AT THE BOTTOM OF THE WASTE HEAT RECOVERY METHODS BECAUSE OF A NUMBER OF PROBLEMS WHICH CAN DEVELOP TO REDUCE ENERGY RECOVERY. THE NEED FOR THE STEAM PRODUCED IS ONE OF THE MORE POSITIVE REASONS FOR THE BOILER; HOWEVER, IT IS SURPRISING THAT ONCE THE STEAM IS PRODUCED, HOW OFTEN THAT NEED DISAPPEARS. IF THE REFINERY DOES NOT HAVE THE CAPACITY TO ADJUST TO THE WASTE STEAM PRODUCTION, THE GENERATED STEAM MAY HAVE TO BE VENTED. A DETAILED STEAM POWER BALANCE SHOULD ALWAYS BE MADE WHEN APPRECIABLE STEAM IS PRODUCED.

THE MOST SERIOUS DRAWBACK ON A STEAM GENERATING WASTE HEAT RECOVERY SYSTEM IS THAT THEY DO NOT GENERATE DESIGN QUANTITIES OF STEAM FOR ONE REASON OR ANOTHER. THIS APPLIES MAINLY TO FURNACE FLUE GAS GENERATING SYSTEMS. WASTE HEAT BOILERS ON OIL STREAMS ARE NORMALLY MUCH MORE EFFICIENT THAN SIMILAR BOILERS ON FURNACE FLUE GAS STREAMS.

WASTE HEAT BOILERS ON FURNACE FLUE GASES HAVE A HISTORY OF NOT PRODUCING THE QUANTITY OF STEAM PREDICTED FOR THE FOLLOWING REASONS:

1. BYPASSING OR RECIRCULATION OF THE FLUE GASES WITHIN THE SYSTEM. WHERE MULTIPLE DUCTING IS EMPLOYED, BALANCING OF FLOW RATES THROUGH THE SYSTEMS IS DIFFICULT. THIS CAN ALSO BE A PROBLEM ON AIR PREHEATERS.

2. MECHANICAL PROBLEMS (VIBRATIONS) THAT CAN LIMIT THROUGHPUT THROUGH EQUIPMENT. THIS CAN BE CAUSED BY THE FAN DRIVES OR BY THE RESONANCE IN THE STEAM GENERATOR ITSELF.

3. DEPOSITS ON TUBE SURFACES THAT INTERFERE WITH HEAT TRANSFER. THESE DEPOSITS CAN BE VERY TENACIOUS AND DIFFICULT TO REMOVE.

4. OPERATORS HAVE A RESISTANCE TO CLOSE DAMPERS ON A FIRED PIECE OF EQUIPMENT BECAUSE OF SAFETY REASONS. JUST BECAUSE THE ECONOMICS OF A PROCESS INDICATES THE DAMPER SHOULD BE CLOSED, DOES NOT MEAN THAT IS THE WAY THE EQUIPMENT IS OPERATED ON THE UNIT.

OTHER WASTE HEAT RECOVERY METHODS

THE WASTE HEAT RECOVERY SYSTEMS PREVIOUSLY MENTIONED ARE EMPLOYED GENERALLY IN REFINERY ENERGY CONSERVATION PROGRAMS. OTHER SCHEMES HAVE BEEN INVESTIGATED IN DETAIL, YET TO DATE HAVE NOT BEEN WIDELY ACCEPTED. THESE MOSTLY INVOLVE RECOVERY OF LOW LEVEL HEAT. PAYOUTS ON THESE FACILITIES ARE DIFFICULT TO OBTAIN.

1. RANKING CYCLE WORK RECOVERY.

2. ABSORPTION REFRIGERATION--THIS IS GETTING CLOSE WITH LITHIUM BROMIDE REFRIGERATION FOR LEAN OIL SYSTEMS.

3. VAPOR RECOMPRESSION.

4. HEAT PUMP APPLICATIONS.

5. COGENERATION (ESPECIALLY LOW LEVEL HEAT USE).

THIS GIVES YOU A VIEW OF HOW AN OPERATOR LOOKS AT WASTE HEAT RECOVERY. ENERGY CONSERVATION IS NOT A CRISIS ITEM IN REFINERY OPERATION. IF SOME DEBACLE OCCURS TO THE WASTE HEAT RECOVERY SYSTEM, MORE OFTEN THAN NOT IT REMAINS BYPASSED AND UNREPAIRED FOR SOME PERIOD OF TIME. THIS INCLUDES SOME MAJOR HEAT RECOVERY EQUIPMENT SUCH AS CO BOILERS. THE SHUTDOWN OF A

CAT CRACKING UNIT AT A CRITICAL TIME, OUT WEIGHS THE ADDED FUEL COSTS THAT THE FAILED CO BOILER MAY CAUSE. THEREFORE, THE RELIABILITY FACTOR ON THE WASTE HEAT RECOVERY EQUIPMENT BECOMES VERY IMPORTANT. WHEN PIECES OF WASTE HEAT RECOVERY EQUIPMENT ARE NOT OPERATING, THE ENERGY CONSUMPTION OF A REFINERY SKY ROCKETS.

PHILOSOPHY

A BASIC PHILOSOPHY SHOULD BE DEVELOPED WHEN EVALUATING A WASTE HEAT RECOVERY SYSTEM. THE SYSTEM MUST WORK AS DESIGNED IN CIRCUMSTANCES TO BE ENCOUNTERED DURING THE USEFUL LIFE OF THE EQUIPMENT.

OPERATING EXPERIENCE ON A NUMBER OF DIFFERENT TYPES OF INSTALLATIONS SHOW THE FOLLOWING TYPE EQUIPMENT TO BE PREFERRED.

1. FACILITIES THAT REDUCE FUEL INPUT TO THE SYSTEM RATHER THAN STEAM GENERATION EQUIPMENT.

2. RELIABLE EQUIPMENT WITH MINIMAL MOVING PARTS TO INCREASE ENERGY SAVING ONSTREAM TIME.

3. EQUIPMENT WHEN RETROFIT IS ADAPTABLE TO ORIGINAL UNIT DESIGN. THE DESIGNER MUST INVOLVE THE OPERATING PEOPLE TO THE CHANGE IN EQUIPMENT. A COMMITMENT TO RUN THE

EQUIPMENT FOR ENERGY SAVINGS MUST BE MADE RECOGNIZING PROBLEMS WITH MANPOWER, SAFETY, AND OTHER CONSIDERATIONS.

A. LARGE MULTIPLE PARALLEL DUCT RUNS SHOULD BE AVOIDED BECAUSE OF FLOW BALANCING PROBLEMS.

B. DAMPERS WHICH SEPARATE COLD AND HOT GAS STREAMS ARE TO BE AVOIDED OR MINIMIZED AS THEY OFTEN BECOME INOPERABLE.

THE REASONING FOR THIS PHILOSOPHY COMES FROM PRACTICAL EXPERIENCE RATHER THAN THE THEORETICAL DESIGN BASIS. OFTEN THE REAL WORLD OPERATIONS ARE DIFFERENT THAN THOSE DEVELOPED ON THE DESIGNER'S DESK. THIS IS NOT TO CAST ASPERSIONS ON THE DESIGNER. THE DESIGNED SYSTEMS ARE BASED ON WORKABLE EQUIPMENT AND REFINERY OPERATIONS AS SPECIFIED. CERTAIN SITUATIONS CREATE OPPORTUNITIES FOR THIS EQUIPMENT TO MALFUNCTION. THE POSSIBILITY OF THESE ABNORMAL SITUATIONS MUST BE EVALUATED WHEN NEW OR REVAMP ENERGY REDUCTION PROJECTS ARE CONSIDERED.

RETURN ON INVESTMENT IS THE CRITERIA FOR CAPITAL SPENDING. MAKE SURE WHEN SPENDING MONEY FOR WASTE HEAT RECOVERY SYSTEMS, THE MONETARY RISK ASSOCIATED WITH CERTAIN TYPES OF EQUIPMENT IS MINIMIZED.

GOOD ENERGY PROJECTS IN A REFINERY ARE IMPLEMENTED ON THE BASIS OF THIS RETURN ON INVESTMENT. THE BEST ARE IMPLEMENTED FIRST. MANY REFINERIES ARE NOW GETTING TO PROJECTS THAT MAY HAVE A MARGINAL RETURN. THESE PROJECTS SHOULD BE REVIEWED CRITICALLY TO BE SURE ALL PROJECTED ENERGY SAVED IS REAL AND THAT A PORTION DOES NOT DISAPPEAR DUE TO UNFORESEEN CONDITIONS.

A81730,1C

Waste Heat Recovery Equipment

1. Additional or Optimized Heat Exchange Surface
2. Additional Furnace Convective Area
3. Furnace Combustion Air Preheaters
4. Furnace Waste Heat Boilers
5. Low Level Energy Recovery

Unit Energy Audit

1. Product Steams to Cooling Air or Water, NMT 250°F

2. Furnace Flue Gas to Atmosphere, NMT 400°F

3. Furnace Excess Air, NMT 20%

4. Radiant Heat Surfaces, NMT 150°F

Furnace Waste Heat Boiler Problems

1. Bypassing or Recirculation
2. Mechanical Problems (Vibrations)
3. Deposits on Tubes
4. Operator Resistance to Efficient Operation

Other Waste Heat Recovery Methods

1. Rankine Cycle Work Recovery
2. Absorption Refrigeration
3. Vapor Recompression
4. Heat Pump Application
5. Cogeneration

Exxon Engineering **Petroleum Department**

EXXON RESEARCH AND ENGINEERING COMPANY

THE USE OF ENERGY EFFICIENT GUIDELINES FOR REFINERY PROCESS UNITS

(Presented by W. Lockett, Jr. at the Waste Heat Utilization Conference sponsored by the Engineering Foundation Conference, August. 1978)

The cost of energy is now the largest single element of operating expense in petroleum refining and exceeds 50% of the total in some countries Therefore, reducing energy consumption can have a significant impact on a refiner's profit. These circumstances justify an agressive campaign to recover energy and amplify the need for an accurate energy management tool to assess the relative energy efficiency of individual process units or refineries. There also exists an inherent need to conserve energy.

Effective utilization of waste heat in petroleum processing facilities involves three basic elements: one, identifying and quantifying areas of recoverable waste heat; two, assessing and implementing energy conservation projects; and. three having an energy management program that measures utilization of waste heat. In addition to recovering waste heat, opportunities for reducing energy input need to be investigated.

Our initial efforts in energy management were based on monitoring MBtu/barrel of crude processed to measure total refinery energy consumption. It was sometimes ineffective and, often, energy conservation gains were masked by increased inefficiencies in other areas. We then tried a system based on energy consumptions of individual process units. It also proved inadequate because: energy requirements per barrel of feed change as operating severity changes; the factors assumed a standard processing equipment configuration and intensity for associated facilities such as feed preparation and light ends fractionation; and the factors were based on minimum average energy consumption, rather than a lower level of energy required in an energy efficient operation.

Other energy utilization monitoring systems using MBtu/B were found to measure energy consumption and not energy performance efficiency Also, the other energy monitoring systems did not fully recognize the differences in processing complexity from location to location nor the differences in severity of operation over time at the same location.

We then developed a new approach to energy management based on energy efficient guidelines. The new system also identifies where energy is being wasted. I will cover the principles incorporated in this approach as you may find them applicable to other process industries.

Energy Efficient Guidelines for Process Units

I would like to share Exxon's experience with the energy efficient guidelines for process units. Basically, the following four areas will be covered: our views on the need for energy management; how we developed the energy efficient concept; application of the Energy Guideline Factor (EGF) system; and our experience with the system.

Need for Energy Management

Five objectives for an effective energy management tool are:

- A reasonable accurate energy measurement system is required for analyzing the overall energy utilization performance.
- The system should identify and quantify areas where energy is being wasted.
- Progress in energy conservation activities should be easily evaluated.
- A means should be provided for establishing and stewarding against ENCON goals.
- A rational system is needed to stimulate interest in conserving energy.

Development of Energy Efficient Concept

Considerable work was undertaken to characterize an energy efficient process unit. Energy efficient criteria were established for recovering energy from furnace stacks and from hot process streams as well as for reduced energy input based on best proven technology. Also, a high degree of integration was assumed between processing units. We then adjusted several Exxon design specifications for the same process unit to meet the energy efficient criteria. The resultant energy consumption was divided by feed rate and called an Energy Guideline Factor or EGF. The factor is expressed as Fuel Oil Equivalent Barrels per Barrels of Feed. An FOEB is valued at 6.05 MBtu.

Since Exxon has designed a large number of projects, we were able to establish a meaningful correlation of key variables affecting energy consumption. This is demonstrated in Figure I which shows energy, in FOEB/B, increasing with increasing severity of operation. As you may know, a reforming unit that produces a higher octane gasoline operates at higher severity and requires more energy.

Correlations were developed for over 40 process units. They represent 95-100% of most refineries' total energy consumption. Separate energy factors were established for each type of light ends fractionator. This provides building blocks for each refiner to define the guideline energy for the actual processing facilities being operated. Guideline energy for a process unit is simply determined by multiplying feed barrels times the Energy Guideline Factor. To make the system easy to use, the factors are related to routine processing conditions (feed type, octane, reflux ratio, etc.). As you might expect, the guideline correlation development required a substantial engineering effort. It took over a year to complete.

Application of EGF System

The resultant EGF system provided an energy management tool that has flexibility to vary guideline energy with operating severity and changing throughputs for each process unit. It provides a common energy efficient base for all types of operation. Comparisons can be made between individual process units or, by summation, complete refineries. This is demonstrated in Table I. For each process unit, the feed barrels are multiplied by the EGF to obtain guideline energy. The difference between actual and guideline energies is the potential recoverable energy.

By dividing actual energy consumed by guideline energy, the percent of guideline or POG is ascertained. It is easy to identify which processes are wasting significant quantities of energy. Process unit A is the major offender. Over a period of time, progress in the energy conservation program can be monitored as shown in Figure II. This refinery reduced energy consumption from 160 to 129 POG, which is equivalent to a 19% reduction.

The results of an energy loss analysis are shown in the simplified summary in Table II. For each area in this pipestill, the delta energy between actual condition and the energy efficient criteria is determined. The furnace stack temperature is 450°F above the energy efficient operation and this is equivalent to 195 FOEB/D of energy that is potentially recoverable from wasted energy in the unit. You will note that the summary provides an overview of where and how much energy is being wasted.

In planning studies and preparation of process design specifications, the EGF system can be used to predict energy consumption for alternate operations and configurations. The EGF system can also be used to provide a better basis for forecasting and stewarding of energy conservation goals.

Over Two Years Experience with EGF System

The last Vu-graph summarizes Exxon's experience with the EGF energy management system. It was easily adopted by our worldwide affiliates, who process a wide range of crude and product slates in many types of refining facilities.

Energy loss analysis has been conducted for most of the process units and this has resulted in uncovering ENCON projects that might have otherwise been overlooked. For example, at one location, lower level heat is now being collected from several process units by a recirculating stream and used to heat tanks and reboil a light ends tower.

Having identified and quantified the potential recoverable energy in the process units, one can develop a matrix that indicates heat source temperature level, anticipated project, potential return on investment, and schedule to recover energy. This type of matrix aids in establishing priorities for implementation of projects. It might be of interest to note that implementing projects involving additional heat exchange or new control systems generally have higher payouts and can be brought to fruition sooner than furnace air preheater projects.

For stewarding energy conservation, we can now develop fairly accurate energy reconciliations. This provides a higher level of confidence in establishing energy conservation goals for the next year and some time in the future (3-5 years). These are goals that are set by the engineers based on identified projects They provide management with a good picture of what can be expected.

All new process unit design projects incorporate the energy efficient guideline principles. Also, the EGF system is used to assess the impact of the new facilities being added to an existing refinery.

The EGF system and our energy conservation guide, a compilation of design, operating and maintenance guidelines for reducing energy consumption in processing equipment, has helped Exxon affiliates to be leaders in percent reduction of energy consumption with time as well as in using low energy per barrel processed. It should be noted that the EGF system has been in operation for over two years and has met the test of time because it measures energy performance efficiency, not just energy performance.

In closing, I trust that the principles incorporated in our energy management system based on the energy efficient approach may find application in other process industries. A similarly structured approach can result in an effective energy management tool.

W. LOCKETT, JR.

WL:cal

Figure 1

ENERGY REQUIRED VS SEVERITY

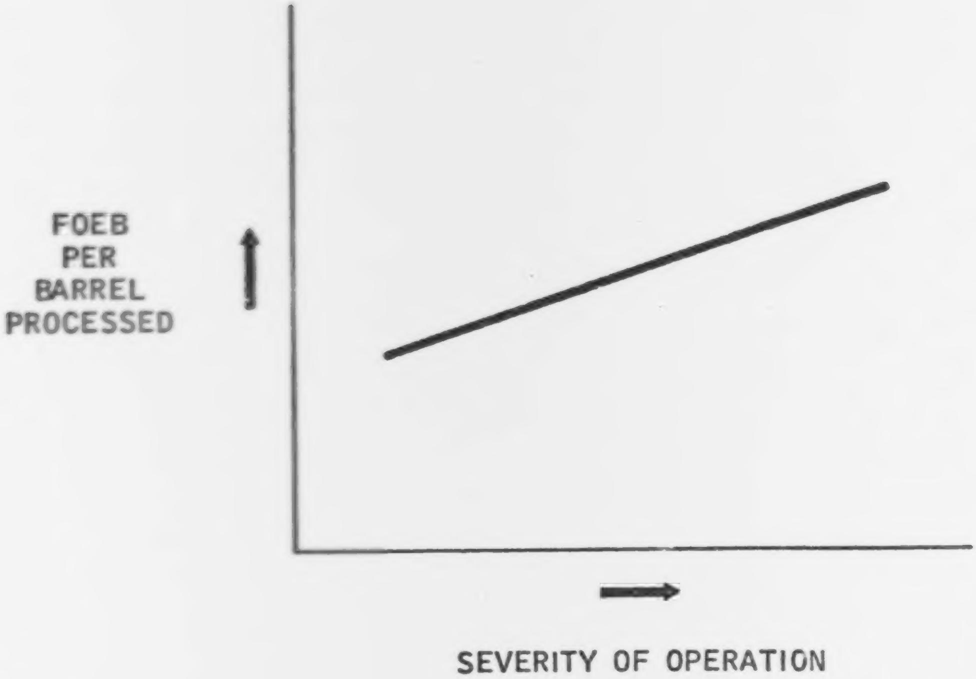


Figure II

MONITORING ENERGY CONSERVATION PROGRESS

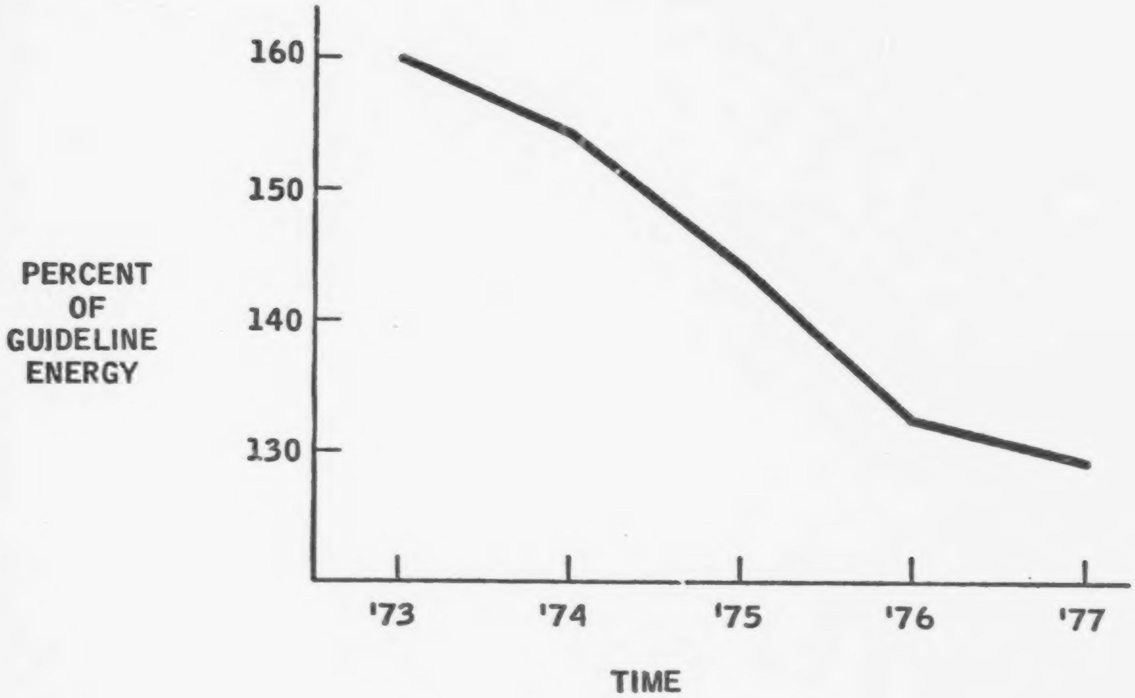


TABLE I

ENERGY EFFICIENT PERFORMANCE OF PROCESS UNITS

			ENERGY, FOEB/D			
UNIT*	B/SD k	EGF %	GUIDELINE	ACTUAL	RECOVERABLE	POG
A	100	1.2	1200	1800	600	150
B	20	1.0	200	220	20	110
C	10	5.0	500	700	200	140
		TOTALS	1900	2720	820	

REFINERY PERCENT OF GUIDELINE

$$\text{POG} = (100)\,\frac{2720}{1900} = 143$$

*INCLUDES FEED PREP. AND LT. ENDS

TABLE II
ENERGY LOSS ANALYSIS RESULTS

ATMOSPHERIC PIPESTILL	DELTA: ACTUAL VS CRITERIA	ENERGY WASTED FOEB/D
FURNACE STACK TEMP	450°F	195
PROCESS STREAM TEMPS.		
• OVERHEAD	75°F	258
• SIDESTREAM	100°F	35
• BOTTOMS	120°F	69
STRIPPING STEAM	20 K LBS/H	125
POWER	290 KWH	10
		692

NEED FOR ENERGY MANAGEMENT

- ANALYZE OVERALL ENERGY UTILIZATION PERFORMANCE
- HIGHLIGHT WHERE ENERGY IS BEING WASTED
- EVALUATE PROGRESS IN ENERGY CONSERVATION PROGRAM
- ESTABLISH AND STEWARD AGAINST A GOAL
- STIMULATE INTEREST IN CONSERVING ENERGY

E-8-78-6

ENERGY EFFICIENT GUIDELINES FOR PROCESS UNITS

- NEED FOR ENERGY MANAGEMENT
- DEVELOPMENT OF ENERGY EFFICIENT CONCEPT
- APPLICATION OF EGF SYSTEM
- EXPERIENCE WITH GUIDELINE SYSTEM

E-8-78-5

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DEVELOPMENT OF ENERGY EFFICIENT CONCEPT

- BASE ENERGY NEEDS ON ENERGY EFFICIENT DESIGNS
- SEVERAL DESIGNS GENERATED TO CORRELATE ENERGY WITH SEVERITY (FIG. 1)
- CORRELATIONS COVER OVER 40 PROCESS UNITS AND FRACTIONATION TOWERS
- GUIDELINE ENERGY = FEED RATE x ENERGY GUIDELINE FACTOR (EGF)
- USE EFFICIENT CRITERIA TO DEFINE WHERE ENERGY CAN BE REDUCED
- HAVE SIMPLE EGF RELATED TO ROUTINE PROCESSING CONDITIONS

E-8-78-8

APPLICATION OF EGF SYSTEM

- PROVIDE ENERGY EFFICIENT BASE THAT VARIES WITH SEVERITY
- CAN COMPARE INDIVIDUAL PROCESS UNITS AND, BY SUMMATION, REFINERIES
- PERCENT OF GUIDELINE USED TO MONITOR ENCON PROGRESS (FIG. 2)
- IDENTIFY PROCESSES THAT ARE WASTING SIGNIFICANT ENERGY
- CONDUCT LOSS ANALYSIS TO QUANTIFY RECOVERABLE ENERGY (TABLE)
- PREDICT ENERGY CONSUMPTION FOR ALTERNATE OPERATIONS/ CONFIGURATIONS
- PERMIT BETTER FORECASTING/STEWARDING OF ENCON GOALS

E-8-78-9

OVER TWO YEARS EXPERIENCE WITH EGF SYSTEM

- EASILY ADOPTED BY WORLDWIDE AFFILIATES (WIDE RANGE OF CRUDE/PRODUCT SLATES)
- LOSS ANALYSIS IDENTIFIED ATTRACTIVE ENCON PROJECTS THAT OTHERWISE WOULD HAVE BEEN OVERLOOKED
- EFFECTIVE IN DEVELOPING/ESTABLISHING PRIORITIES FOR ENCON PROJECTS
- MEANINGFUL ENERGY RECONCILIATIONS NOW POSSIBLE
- USED EXTENSIVELY TO ESTABLISH AND STEWARD FUTURE ENCON GOALS
- ASSESS ENERGY IMPACT FOR NEW PROCESSING FACILITIES
- HAS ASSISTED REFINERIES IN ACHIEVING SIGNIFICANT IMPROVEMENTS
    - LEADERS IN PERCENT IMPROVEMENT
    - LEADERS IN LOW ENERGY PER BARREL PROCESSED
- SYSTEM HAS MET TEST OF TIME

E-8-78-10

LOCATING HEAT RECOVERY OPPORTUNITIES

Presented by Alfred F. Waterland, PE
at the Engineering Foundation Conference on
WASTE HEAT UTILIZATION at Henniker, New Hampshire
August 14-18, 1978

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LOCATING HEAT RECOVERY OPPORTUNITIES

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Before trying to explain how to locate heat recovery opportunities perhaps a few words on the philosophy of Waste Heat Recovery and Energy Conservation would be in order. I have claimed at times that there is no such thing as "Waste Heat," only "Heat that is Wasted." This is admittedly a play on words, but when we consider that energy has value only by virtue of its level above an available heat sink and when we consider that virtually all energy expended by industry is "wasted" to the atmosphere or to a cooling water system, then some of the implications of this statement are more apparent.

Heat Recovery is a very important part of energy conservation. The basic energy conservation objectives are two-fold. First of all, we want to devise ways in which the job can be done with less net energy input. Secondly, we want to get the greatest practical benefit from every unit of energy that must be expended. The second objective is really the conservation of energy level. We want to introduce energy at the highest practical level, then only grudgingly permit this energy to degrade to the lowest possible final rejection level. We want to avoid high energy rejection levels and this really is what we mean by "Heat Recovery." High temperature heat rejections are potential sources of recovered heat. If we can pick up the energy at a high rejection level and make it perform a useful function as it degrades to a lower rejection level, this is Heat Recovery.

There are three ingredients of practical heat recovery. These are shown on Slide No. 1. Briefly, these are a Source, a Use, and a heat transfer Means.

In surveying an industrial plant for heat recovery opportunities, it is necessary to classify heat rejections against an appropriate set of criteria. Slide No. 2 lists some of the source evaluation criteria that have been used for a specific plant study. The level and quantity criteria are not absolute and do not apply to all situations. For example, a very large quantity of available heat could lead to the exceptance of a lower level or vice versa.

Slide No. 3 lists some of the evaluation criteria applicable to recovered heat uses. Slides 4, 5, and 6 illustrate a range of systematic methods used in locating, evaluating, and classifying heat recovery opportunities. I should point out that the process is not as easy and straight-forward as it might first appear. Judgment and experience, together with a developed ability to be thorough without becoming bogged down in superfluous detail are essential.

Another factor that often impedes waste heat recovery progress is failure to recognize that economic incentives have increased so rapidly that action now is virtually mandatory. So many times when a waste heat recovery proposal is discussed, we hear statements like: "We looked at that five years ago and it isn't justified;" or "They tried something like that at one of our other plants but it got to be such a maintenance headache that they wound up leaving it on by-pass permanently." It is an unfortunate attribute of human nature that people tend to resist change. Yet, if energy is to be conserved, and if waste heat is to be recovered, there must be changes -- our attitudes and preconceived notions must change, our methods of doing things must change, and there must be changes in the physical configuration, extent and types of industrial systems and equipment. Increased economic incentive to conserve energy and technological advances are very rapidly whittling away at the valid objections to heat recovery schemes. The thing that still must be overcome is our reluctance to venture forth on heat recovery projects until several other firms have done the same thing successfully.

Slide No. 7 lists some of the devices that can be used to get heat from the recovery source that we have located and deliver it to the matching use that we have managed to find.

The first item (direct mixing, contact condensing, and recycling) can be accomplished in certain cases with no heat transfer devices (in the usual sense). For example, the main value of steam condensate is the sensible heat that it contains. The condensate can be piped back to the boiler plant and mixed directly with the required amount of make-up water, thereby, recovering the heat that the condensate contains. Slide No. 8 shows a device for recovering heat from flash vapors using a contact condenser. Sometimes we find that it is possible and practical to recover heat simply by recycling all or part of a hot exhaust stream back into the device from which it came. If a dryer, for example, is operated with 100% recycle of the drying medium, the drying medium will become superheated steam which has some marked advantages over air.

Conventional heat exchangers will continue to be the mainstay of heat recovery. Heat exchangers of many types are used by themselves as the primary heat recovery device or as major components of sophisticated and ingenious heat recovery systems. Slide No. 9 shows a scheme which is really a combination of direct contact heat transfer and heat transfer through a conventional heat exchanger. Hot incinerator gases at 1800°F were being cooled by contact with quench water before passing into an elaborate chemical absorption flue gas clean-up system. Before heat recovery was added, the cooling effect was accomplished by permitting the circulating quench water to vaporize. The scheme shown in this slide was practical because there were appropriate low level heat uses in the 120°F to 140°F range. The scheme shown is far better than throwing all of the heat away as was done previously, but it falls far short of the ultimate potential because it still permits high level heat (1800°F) to be degraded without benefit to the 200°F circulating quench water level.

Slide No. 10 illustrates the liquid "run-around" system in an application recovering heat from the exhaust of a dryer and using this heat to preheat make-up air to the dryer and building. In this case, the air from which the heat is recovered is very moist. Similar systems will work well on dry air or liquid streams. The important point here is that the amount of heat recovered from moist air can be greatly increased if a portion of the moisture is actually condensed so that both latent heat and sensible heat are recovered.

Slide No. 11 shows the amount of heat recovered when a given quantity of moist air is cooled from 187°F dry bulb and 128°F dew point to various terminal temperatures. Note that the amount of sensible heat recovered from the air in cooling 59° to the dew point is minor compared to the amount recovered by cooling to just 28° below the dew point.

The heat exchanger makes it possible to recover heat. Yet, in most cases it also inherently reduces the amount of benefit that can be realized from this recovered heat. As shown in Slide No. 12, this is because a "drive" or temperature difference must be applied to make the heat flow across the heat transfer surface. The ΔT constitutes a degradation of energy level. The chart shows the losses in energy level occurring in the liquid run-around system illustrated in Slide No. 10. At first it appears that the need for two heat exchangers in this system is a distinct disadvantage and that it would be much better to have the air streams being cooled and heated applied to one heat exchanger. The penalty is not so great, however, when we consider that the limitation is the air-side heat transfer coefficient which is perhaps less than one-tenth the coefficient on the liquid side.

The undesirable degradation in energy level across heat transfer surfaces in heat recovery applications can be reduced by increasing the amount of surface used. This, of course, also increases the project cost but often not in proportion to the increase in heat recovery. For the example liquid run-around heat recovery system, the economics were recalculated using approximately twice the heat exchange surface originally specified. As shown in the table on Slide No. 13, this resulted in a 56% increase in heat recovered for a 21% increase in total project investment cost.

Slide No. 14 shows a unique application of the liquid run-around system for heat recovery. Ammonia stored as a liquid in a refrigerated tank at atmospheric pressure, must be warmed before it can be safely shipped in ordinary steel piping or tank cars. Previously, the ammonia was warmed from -27°F to 35°F using steam. This was an extreme level degradation for the steam whereas this heat use represents a very low level requirement. In the Slide, I show returned cooling tower water as the heat source. Another potential heat source that could effect a double savings, would be to warm the liquid ammonia by providing needed process or air conditioning refrigeration. In the example, "scheduling" criteria ruled this out since the need to warm the ammonia was intermittent.

I have said that the level of a heat recovery source must be high enough with respect to the level of the proposed heat use for a heat recovery scheme to be practical. I did not mean to imply that the source level had to be higher than the use level. One of the most important heat recovery concepts now and for the years ahead is the device known as the "Heat Pump," the "Reverse Rankine Cycle," or the "Vapor Compression System." Actually, all of these are the same thing. Energy level is restored by application of a certain amount of prime energy (shaft work) and the same Btu's are used over and over again. Slide No. 15 shows a typical organic heat pump arrangement applied to heat process water to 180°F using the otherwise rejected heat in cooling water at 130°F. Slide No. 16 shows a single effect evaporator using steam heat in the upper diagram and converted to vapor compression in the lower diagram. In this case the vapor compression evaporation scheme saves approximately $600,000 per year in energy cost.

It should be noted that the practicality of vapor compression or heat pump energy recovery is inversely proportional to the rise in level that must be accomplished. The greater the level separation between source and use, the more compression machinery is required and the more high value prime energy must be expended. Even so, the magnitude of the heat recovery opportunity -- the scale factor, if you will -- can within limits overcome a relatively wide temperature range and still be technically feasible and economically practical. Slide No. 17 shows a direct vapor compression heat recovery application that makes 15 psig steam from 180°F hot water.

Sometimes large quantities of low level heat are available for recovery but there are no thermal uses for this heat at a level anywhere near close enough to make direct recovery or even vapor compression practical. This is where the much-discussed low level Rankine cycle or bottoming cycle could have an application. Slide No. 18 shows the same hot process waste water heat source and the same disengaging drum that was shown in Slide No. 17. Instead of compressing the vapor, however, it is expanded instead through a low pressure steam turbine and exhausted at 2" Hg A into a water cooled condenser. This produces shaft work which can be used directly or converted to electricity as shown.

For years it has been common practice in energy conscious industrial plants to generate plant steam at a "power" level then reduce its pressure to the process use level by expanding it through "topping" or "back-pressure" steam turbines. This heat recovery concept was called the topping turbine principal, heat drop power, or by-product work. It is now referred to, very loosely and imprecisely, as "Cogeneration." A somewhat more descriptive modern term is "Dual Energy Use System." This goes under the acronym DEUS. Slide No. 19 is a scheme applied to a forge plant which does DEUS one better. I call this TEUS for triple energy use system. First of all the high temperature furnace atmosphere is used for heating the work pieces. The hot combustion gases then go to the waste heat boiler and combustion air preheater combination. The steam from the boiler is used to drive steam forge hammers and a turbogenerator. The exhaust from the hammers and turbine provides factory space heating, absorption refrigeration for air conditioning, and other thermal heat requirements. Note that the steam condensate is returned so that its heat is not lost. The only heat that is lost is radiation and convection from the forge heating furnace and the flue gas from the air preheater at 300°F. What can we do with the heat in that 300°F flue gas?

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PRACTICAL HEAT RECOVERY REQUIREMENTS

A HIGH ENOUGH LEVEL SOURCE OF ENOUGH ENERGY

A PRACTICAL MEANS OF GETTING THE ENERGY FROM THE SOURCE TO THE USE

A LOW ENOUGH LEVEL VALID USE FOR AN APPROPRIATE QUANTITY OF ENERGY

HEAT RECOVERY
SOURCE EVALUATION CRITERIA

- **LEVEL**

  Liquids & Condensing Vapors, > 200°F

  Air & Gases, > 400°F

  NOTE: Many exceptions depending on specific application conditions.

- **QUANTITY**

  Minimum 1.0 MM Btu/Hr. usually.

  NOTE: Often much less depending on level, other conditions, and what is required. Also, if sources are small, use several.

- **CORROSIVE, DIRTY, DANGEROUS**

  Doesn't necessarily stop you -- must still be considered. There are ways around all problems if the incentive is right.

- **SCHEDULE**

  Waste heat available in the summer doesn't usually help much with winter space heating.

HEAT RECOVERY
USE EVALUATION CRITERIA

- **LEVEL**

  Is it low enough; or will economically excessive, heat transfer facilities be required? What other sources can match?

- **QUANTITY**

  How much heat can be used versus heat available?

- **FOULING, CORROSION, PROCESS HAZARDS**

  Don't neglect these problems; but don't let them stop good projects unnecessarily.

- **SCHEDULE**

  Continuous uses are best, they can even receive heat from intermittent sources with make-up from fuel or steam. Operational integration with source is preferred, but not a requirement.

- **PROXIMITY TO SOURCE**

  The closer the better but there are trade-offs.

SLIDE NO. 4

PRODUCT DISTILLATION COLUMN ENERGY ANALYSIS

Flow Stream	M Lbs/Hr.	MM Btu/Hr.	Temperatures, °F From	To	At
ENERGY INPUTS					
1. 275 psig shell steam to heater E-716; Approx. 210°F in shell, heats 4200 lbs/hr. of 88% prod. feed from 140°F to 347°F (347-140) 0.627x4200 = 0.545 MM Btu/hr. and provides boil-up ht. -	2.94	2.39	-	-	392
2. 150 psig steam to product stripper reboiler -	0.30	0.30	-	-	366
3. 150 psig injection steam to still heater -	0.50	0.50	-	-	366
		3.19			
ENERGY REJECTIONS					
1. Recycle liquid cooler (273-144) 0.576x15.3 = 1.14 -	15.3	1.14	273	144	-
2. Product cooler, T-409 (347-176) 0.576x3209 = 0.32 -	3.20	0.32	347	176	-
3. Vapors to jets, h_{fg} = 1065 -	1.045	1.11	-	-	158
4. Losses -	-	0.62	-	-	-
		3.19			

TABLE 2

PROCESS AREA NO. 4

HEAT REJECTIONS
WITH INITIAL PROCESS TEMPERATURES 200°F OR HIGHER

Heat Exchanger No.	Name	Process Flow Lb/Hr.	°F In	°F Out	Approx. Heat Duty 10^6 Btu/Hr.
250-21	Material A Column Condenser	14320	305	265	2.39
250-28	Heavy Residue Cooler	1000	330	150	0.07
250-31	Finishing Column Bottoms Cooler	5390	364	150	0.58
250-33	Finishing Column Condenser	33790	329	294	5.55
250-40	Material B Stripper Condenser (Chilled Water)	2290	222	64	0.86
260-44	Stripper Bottoms Cooler	6870	222	132	0.32
280-17	1st Stage Material C Condenser	5690	222	110	1.91
280-25	2nd Stage Material C Condenser (Brine Cooled)	660	222	24	0.17
310-94	Material D Condenser	6840	272	120	3.18
340-02	Material D Distillation Column Condenser	11060	232	132	3.09
340-05	Recycle Material E Condenser	613	303	294	0.15
	TOTAL				18.27

SLIDE NO. 5

SLIDE NO. 6

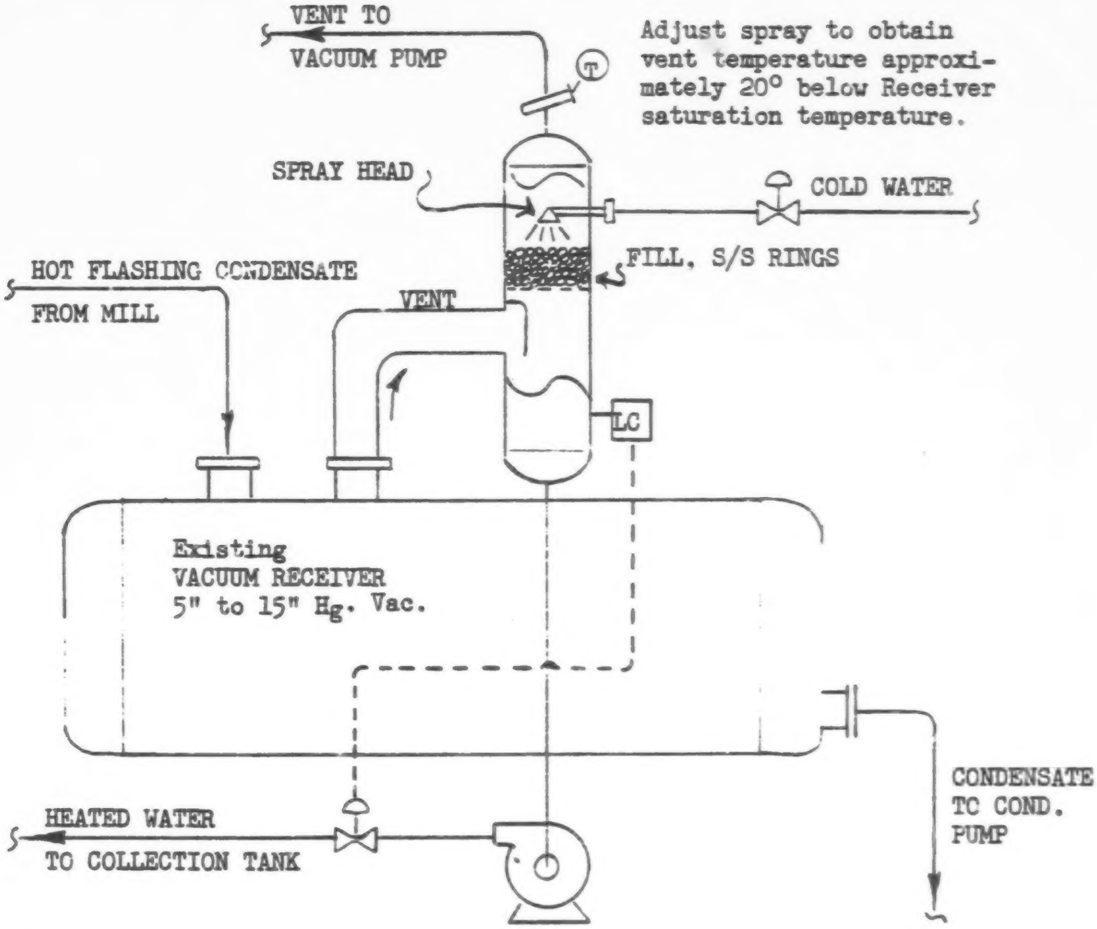
DISTILLATION TRAIN ANALYSIS

	1st Separation Column	2nd Separation Column	Finishing Column
Reboiler Steam Flow - Lbs/Hr.	5196	3836	13657
Feed Flow - Lbs/Hr.	27614	27348	25947
Overhead Vapor Flow - Lbs/Hr.	3979	26772	63180
Reflux Flow - Lbs/Hr.	3682	784	47003
Reflux, % of Overhead Vapor	93	3	75
Feed Temperature, °F	113	338	255
Bottoms Temperature, °F	338	320	354
Overhead Temperature, °F	194	295	315
Reflux Temperature, °F	99	255	284
Column Pressure, mm Hg A.	70	30.5	60.5

HEAT TRANSFER SYSTEMS

- DIRECT MIXING, CONTACT CONDENSING, RECYCLING
- CONVENTIONAL HEAT EXCHANGERS
  - SHELL & TUBE
  - PLATE
  - PLATE COIL
- REGENERATIVE HEAT EXCHANGERS
  - LJUNGSTROM
  - HEAT WHEEL
  - BRICK CHECKER-WORK CHAMBERS
- HEAT PIPE
- LIQUID RUN-AROUND SYSTEMS
- WASTE HEAT BOILERS
  - HOT GASES TO STEAM
  - HOT LIQUID TO STEAM
  - CONDENSING PROCESS VAPORS TO STEAM

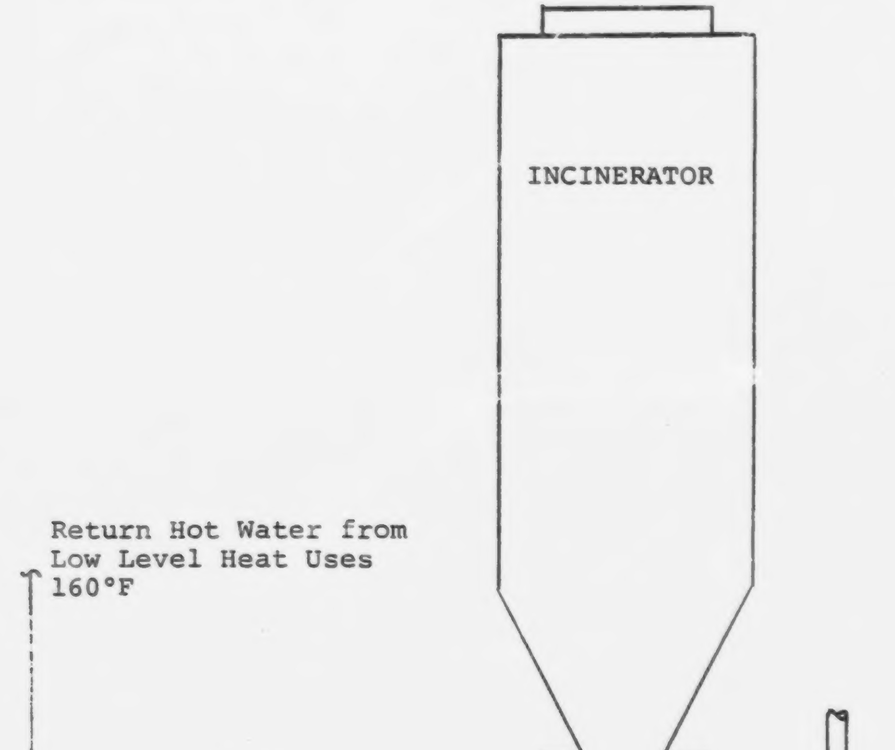
VACUUM RECEIVER HEAT RECOVERY
USING DIRECT CONTACT VENT CONDENSER


VENT TO
VACUUM PUMP
T
SPRAY HEAD
COLD WATER
FILL, S/S RINGS
HOT FLASHING CONDENSATE
FROM MILL
VENT
LC
Existing
VACUUM RECEIVER
5" to 15" Hg. Vac.
CONDENSATE
TC COND.
PUMP
HEATED WATER
TO COLLECTION TANK

Adjust spray to obtain vent temperature approximately 20° below Receiver saturation temperature.

NOTE: 1. Suggest: vent condenser 24" dia. x 6' high.
From Lower Head: Liquid level 1 ft.
Top of Vent Entry 2 ft.
Fill Support Grid 2 ft.
Top of Fill 4 ft.
Spray Head 5 ft.
2. Fill: Commercial S/S Fill or cut 1" rings from 1-1/2" S/S pipe.
3. Vent Condenser does not have to be above receiver.

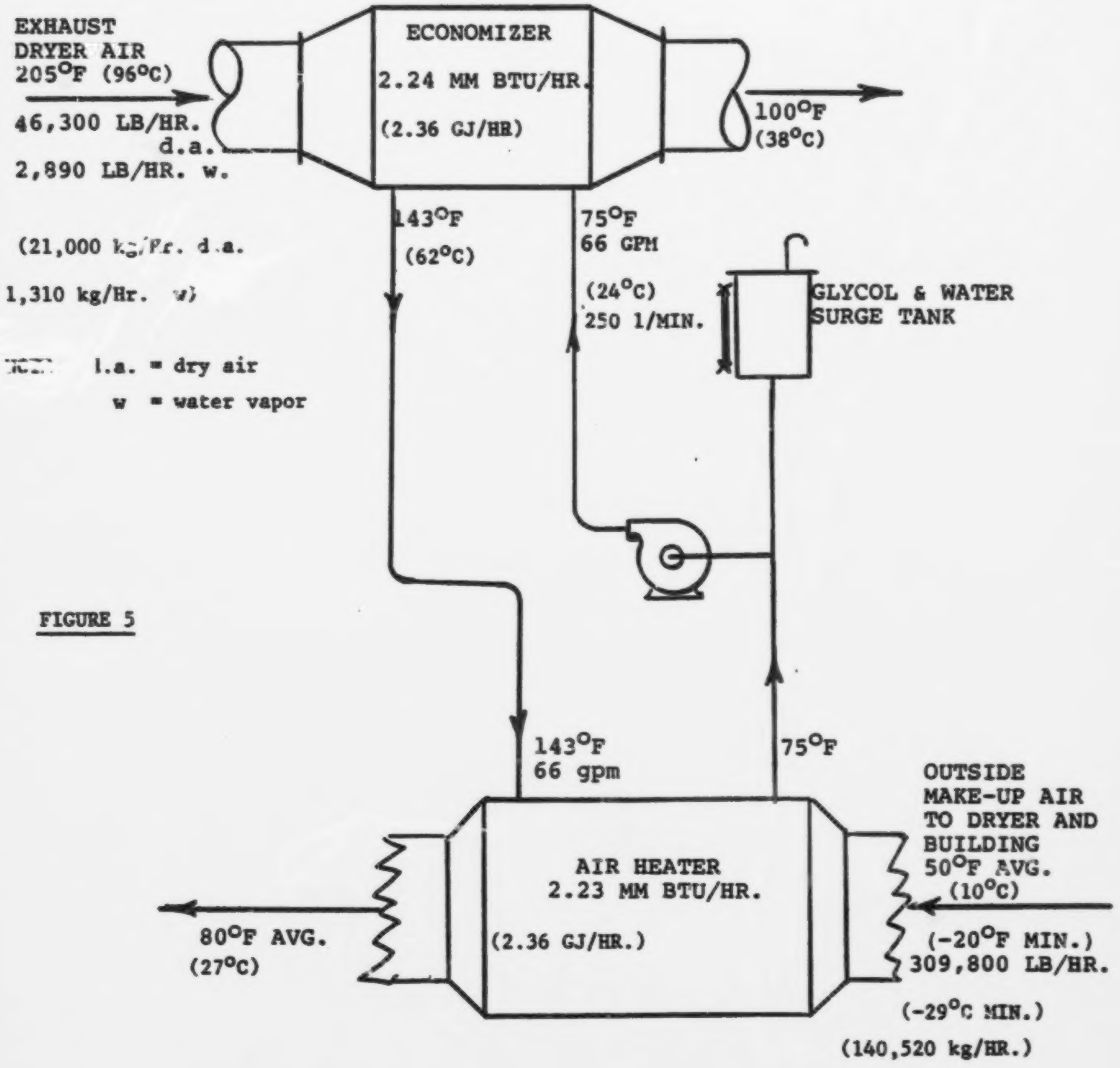
INCINERATOR QUENCH POT HEAT RECOVERY


INCINERATOR
Return Hot Water from Low Level Heat Uses 160°F
180°F
Heat Exchanger
NC
QUENCH POT
Existing Quench Recirculating Pumps
200°F
Hot Water to Low Level Heat Use, 180°F

SLIDE NO. 10

AIR TO AIR
HEAT RECOVERY

LIQUID "RUN-AROUND" SYSTEM



EXHAUST
DRYER AIR
205°F (96°C)

46,300 LB/HR.
d.a.
2,890 LB/HR. w.

(21,000 kg/Hr. d.a.
1,310 kg/Hr. w)

NOTE: d.a. = dry air
w = water vapor

ECONOMIZER
2.24 MM BTU/HR.
(2.36 GJ/HR)

100°F
(38°C)

143°F
(62°C)

75°F
66 GPM
(24°C)
250 l/MIN.

GLYCOL & WATER
SURGE TANK

143°F
66 gpm

75°F

AIR HEATER
2.23 MM BTU/HR.
(2.36 GJ/HR.)

80°F AVG.
(27°C)

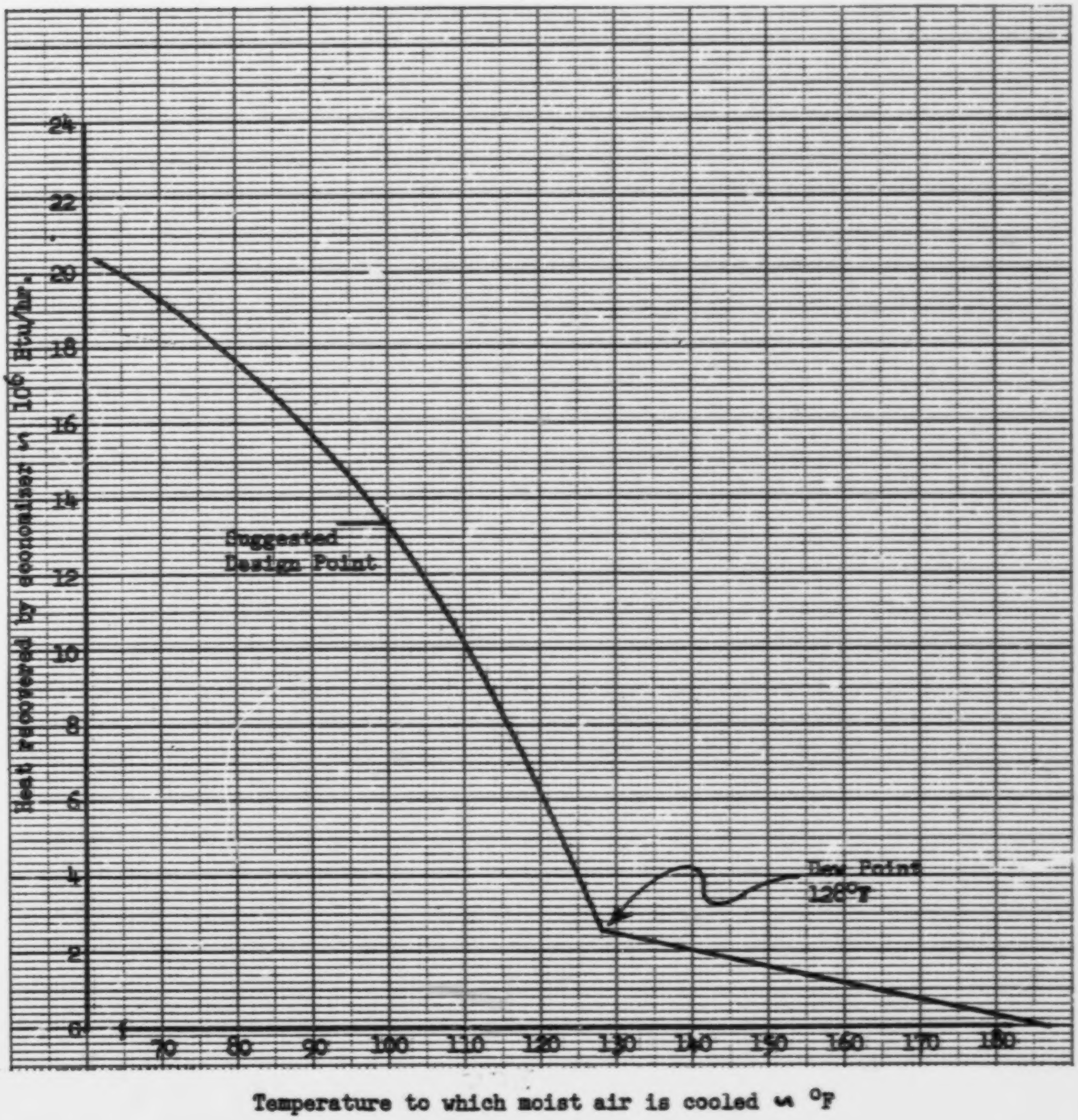
OUTSIDE
MAKE-UP AIR
TO DRYER AND
BUILDING
50°F AVG.
(10°C)
(-20°F MIN.)
309,800 LB/HR.
(-29°C MIN.)
(140,520 kg/HR.)

FIGURE 5

SLIDE NO. 11

PAPER MACHINE ECONOMIZER HEAT RECOVERY FROM MOIST AIR VS. OUTLET AIR TEMPERATURE

EXAMPLE:
154,300 lb/hr. dry air
15,900 lb/hr. water vapor
Humidity ratio 0.103 lb.w/lb.da
Temperature, 187°F db
Dew Point, 128°F



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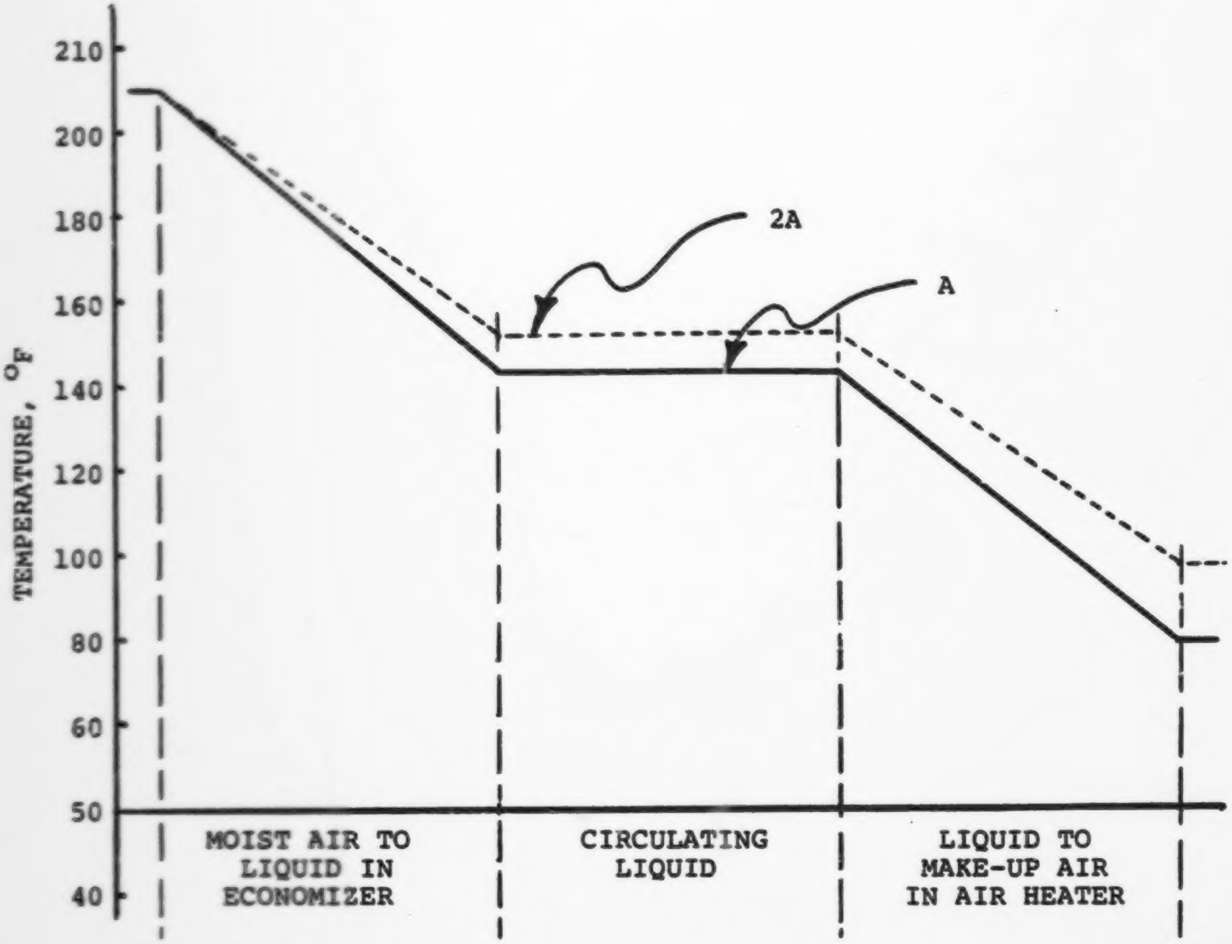
SLIDE NO. 12

ENERGY LEVEL DEGRADATION ACROSS HEAT EXCHANGERS
LIQUID RUN-AROUND HEAT RECOVERY



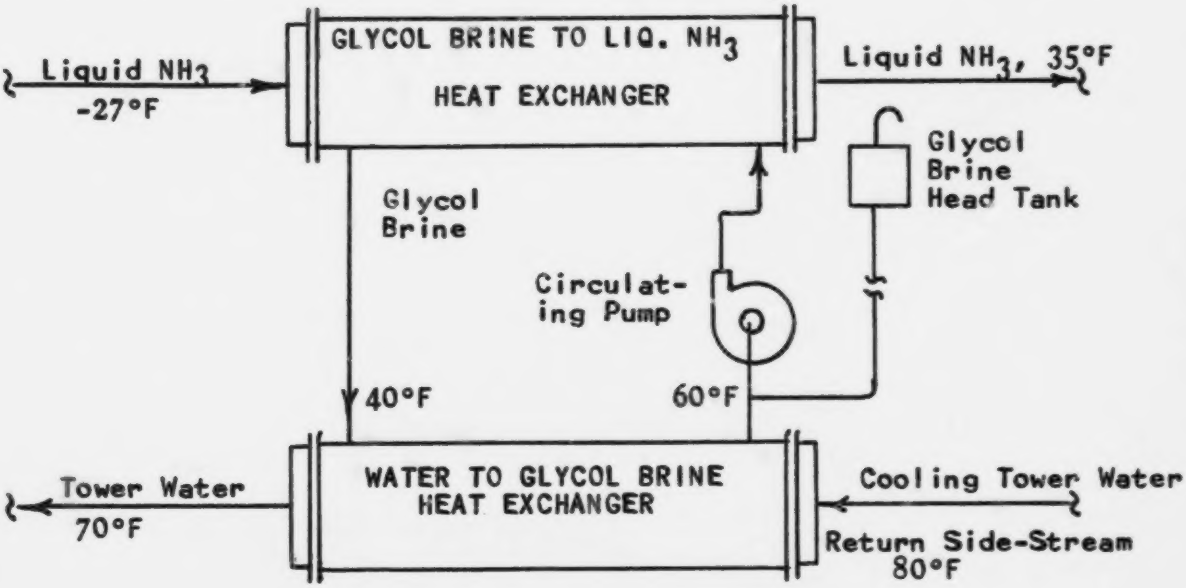
EFFECT OF DOUBLING TOTAL HEAT TRANSFER SURFACE AREA

TOTAL SURFACE AREA	A	2A
HEAT RECOVERED, MM BTU/HR.	2.24	3.49
PROJECT INVESTMENT COST, %	100	121
HEAT RECOVERED, %	100	156

SLIDE NO. 14

REFRIGERATED AMMONIA STORAGE

HEAT RECOVERY FOR WARMING LIQUID AMMONIA



Liquid NH_3 -27°F

GLYCOL BRINE TO LIQ. NH_3 HEAT EXCHANGER

Liquid NH_3, 35°F

Glycol Brine Head Tank

Glycol Brine

Circulating Pump

40°F

60°F

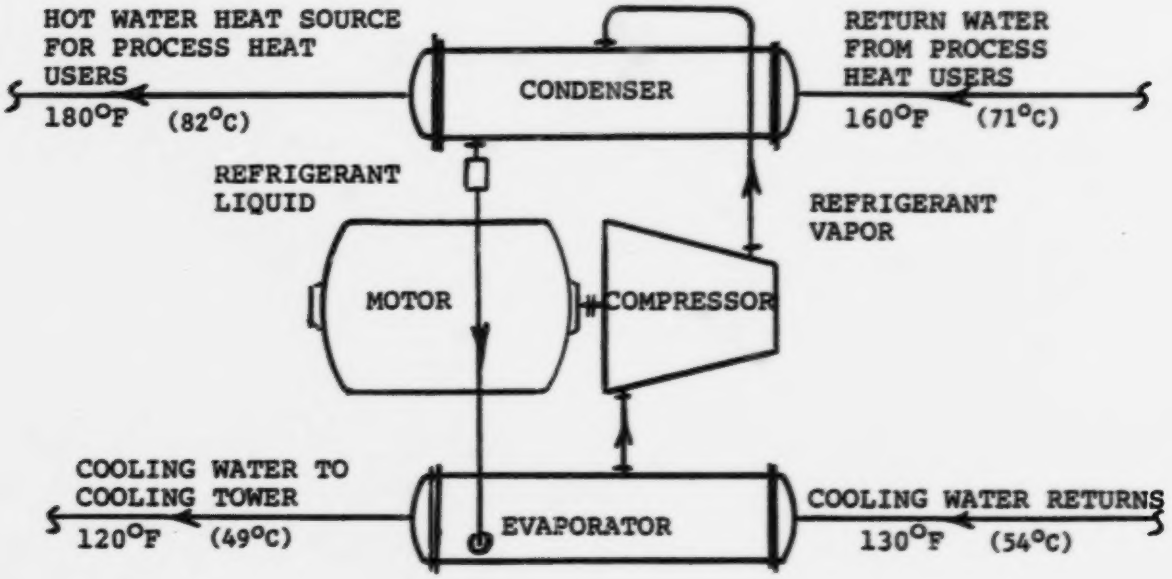
Tower Water 70°F

WATER TO GLYCOL BRINE HEAT EXCHANGER

Cooling Tower Water Return Side-Stream 80°F

APPLICATION OF ORGANIC HEAT PUMP

FIGURE 4



HOT WATER HEAT SOURCE FOR PROCESS HEAT USERS

180°F (82°C)

CONDENSER

RETURN WATER FROM PROCESS HEAT USERS

160°F (71°C)

REFRIGERANT LIQUID

REFRIGERANT VAPOR

MOTOR

COMPRESSOR

COOLING WATER TO COOLING TOWER

120°F (49°C)

EVAPORATOR

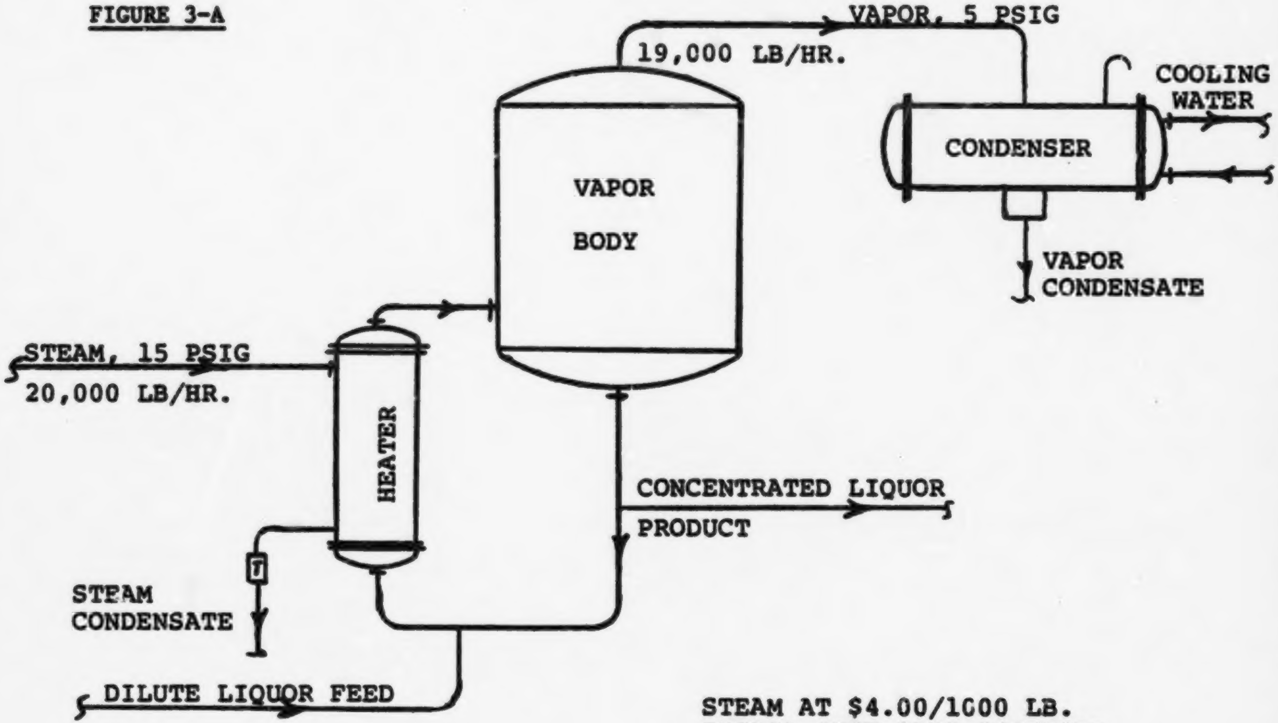
COOLING WATER RETURNS

130°F (54°C)

SLIDE NO. 16

SINGLE EFFECT EVAPORATOR ON STEAM

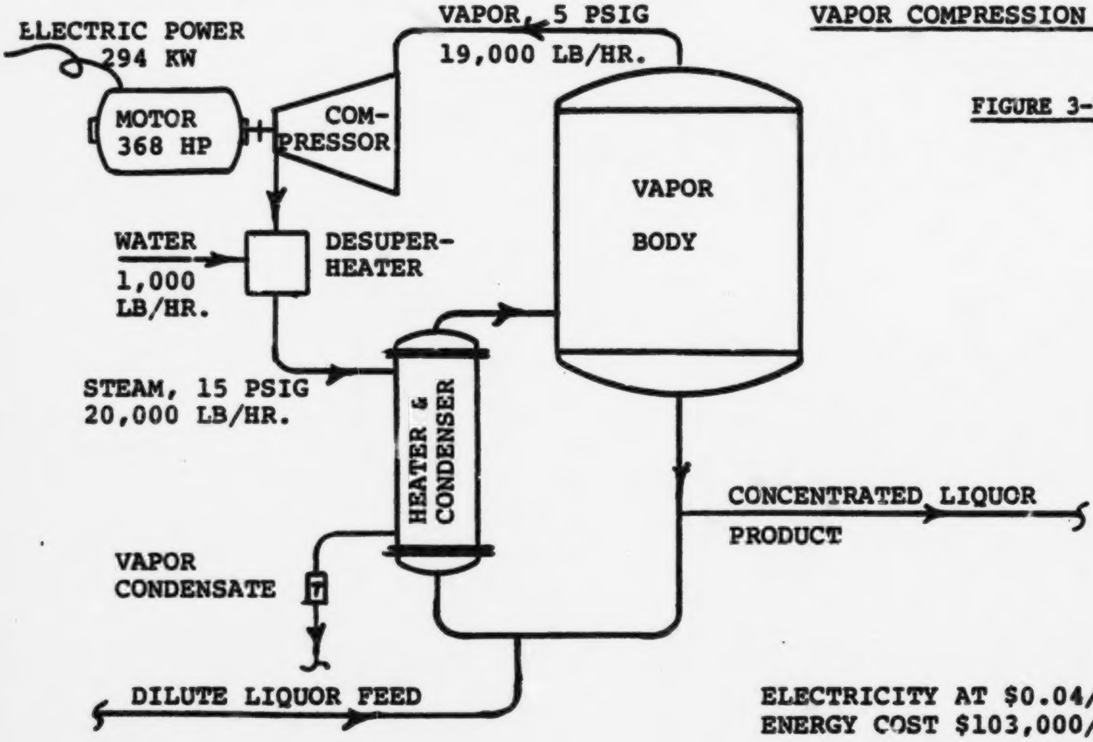
FIGURE 3-A


VAPOR, 5 PSIG
19,000 LB/HR.
COOLING WATER
CONDENSER
VAPOR BODY
VAPOR CONDENSATE
STEAM, 15 PSIG
20,000 LB/HR.
HEATER
CONCENTRATED LIQUOR PRODUCT
STEAM CONDENSATE
DILUTE LIQUOR FEED

STEAM AT $4.00/1000 LB.
ENERGY COST $700,800/YEAR

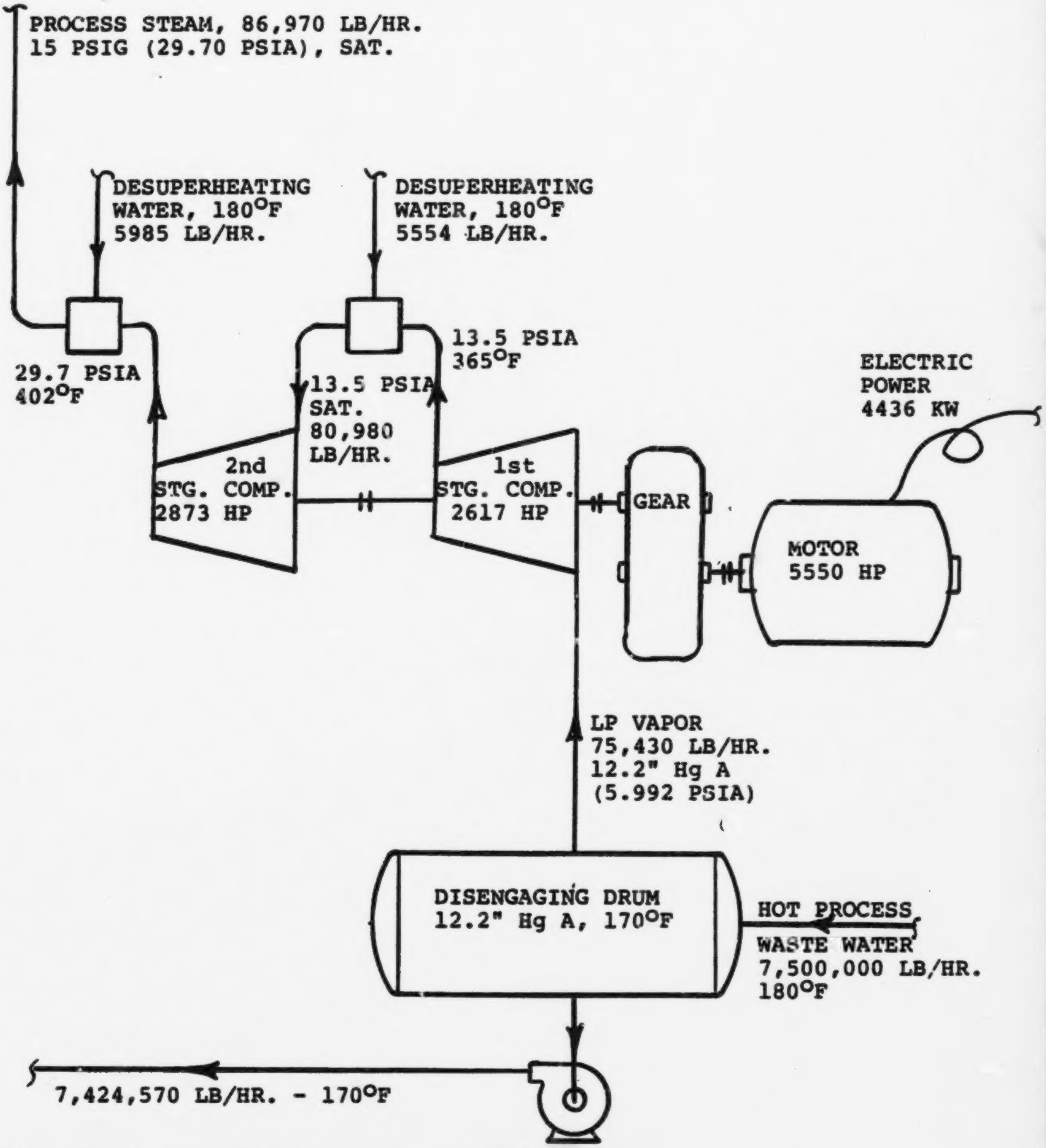
VAPOR COMPRESSION EVAPORATOR

FIGURE 3-B


VAPOR, 5 PSIG
19,000 LB/HR.
ELECTRIC POWER
294 KW
MOTOR
368 HP
COM-PRESSOR
WATER
1,000 LB/HR.
DESUPER-HEATER
VAPOR BODY
STEAM, 15 PSIG
20,000 LB/HR.
HEATER & CONDENSER
CONCENTRATED LIQUOR PRODUCT
VAPOR CONDENSATE
DILUTE LIQUOR FEED

ELECTRICITY AT $0.04/KWH
ENERGY COST $103,000/YEAR

VAPOR COMPRESSION HEAT RECOVERY



PROCESS STEAM, 86,970 LB/HR.
15 PSIG (29.70 PSIA), SAT.

DESUPERHEATING
WATER, 180°F
5985 LB/HR.

DESUPERHEATING
WATER, 180°F
5554 LB/HR.

29.7 PSIA
402°F

13.5 PSIA
SAT.
80,980
LB/HR.

13.5 PSIA
365°F

ELECTRIC
POWER
4436 KW

2nd
STG. COMP.
2873 HP

1st
STG. COMP.
2617 HP

GEAR

MOTOR
5550 HP

LP VAPOR
75,430 LB/HR.
12.2" Hg A
(5.992 PSIA)

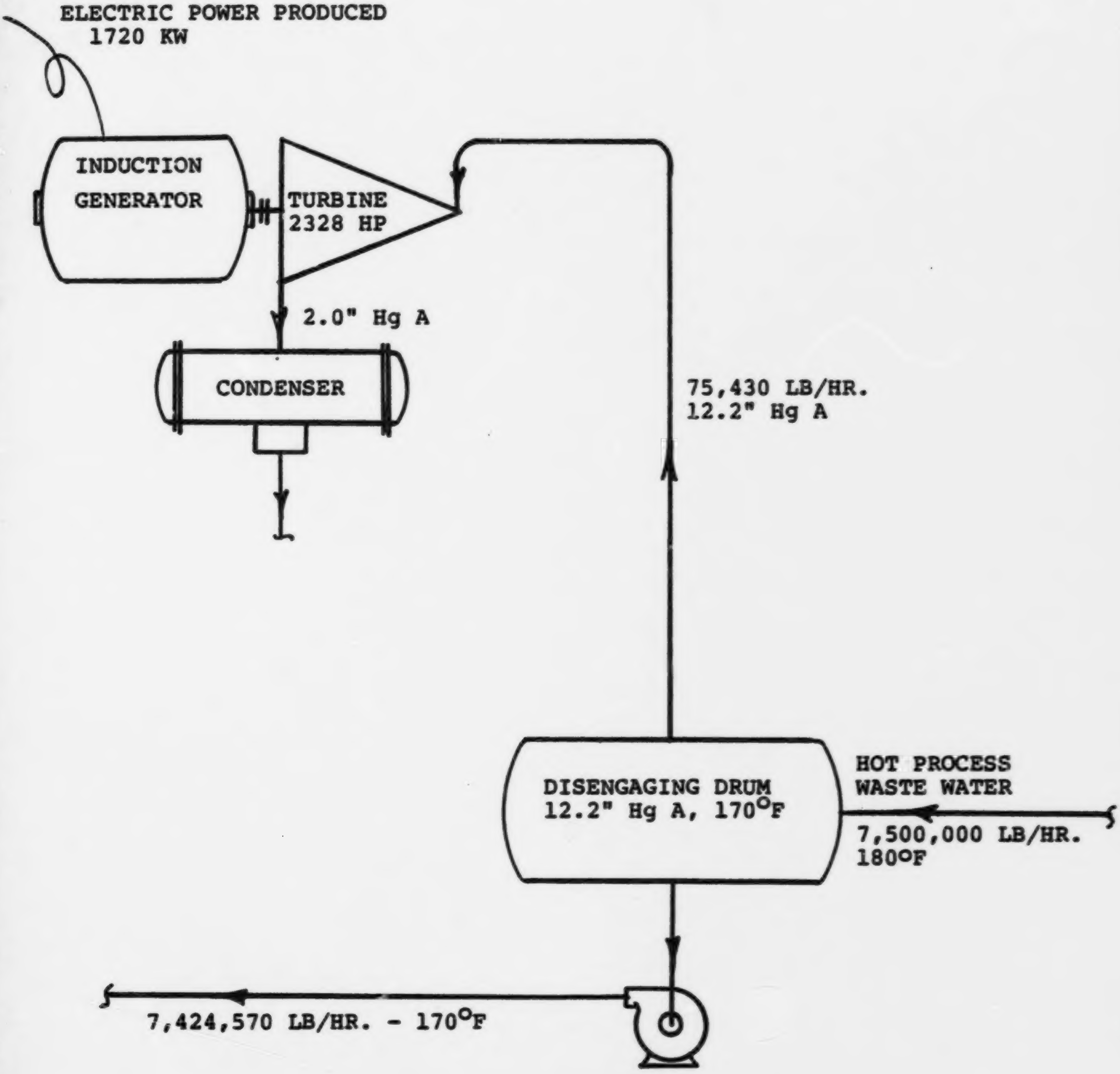
DISENGAGING DRUM
12.2" Hg A, 170°F

HOT PROCESS
WASTE WATER
7,500,000 LB/HR.
180°F

7,424,570 LB/HR. - 170°F

@ $4.00/1000 LB, STEAM IS WORTH	$3,047,300/YEAR
@ $0.04/KWH, ELECTRIC POWER COSTS	$1,554,400/YEAR
SAVINGS	$1,492,900/YEAR

LOW TEMPERATURE RANKINE CYCLE



ELECTRIC POWER PRODUCED
1720 KW

INDUCTION GENERATOR

TURBINE
2328 HP

2.0" Hg A

CONDENSER

75,430 LB/HR.
12.2" Hg A

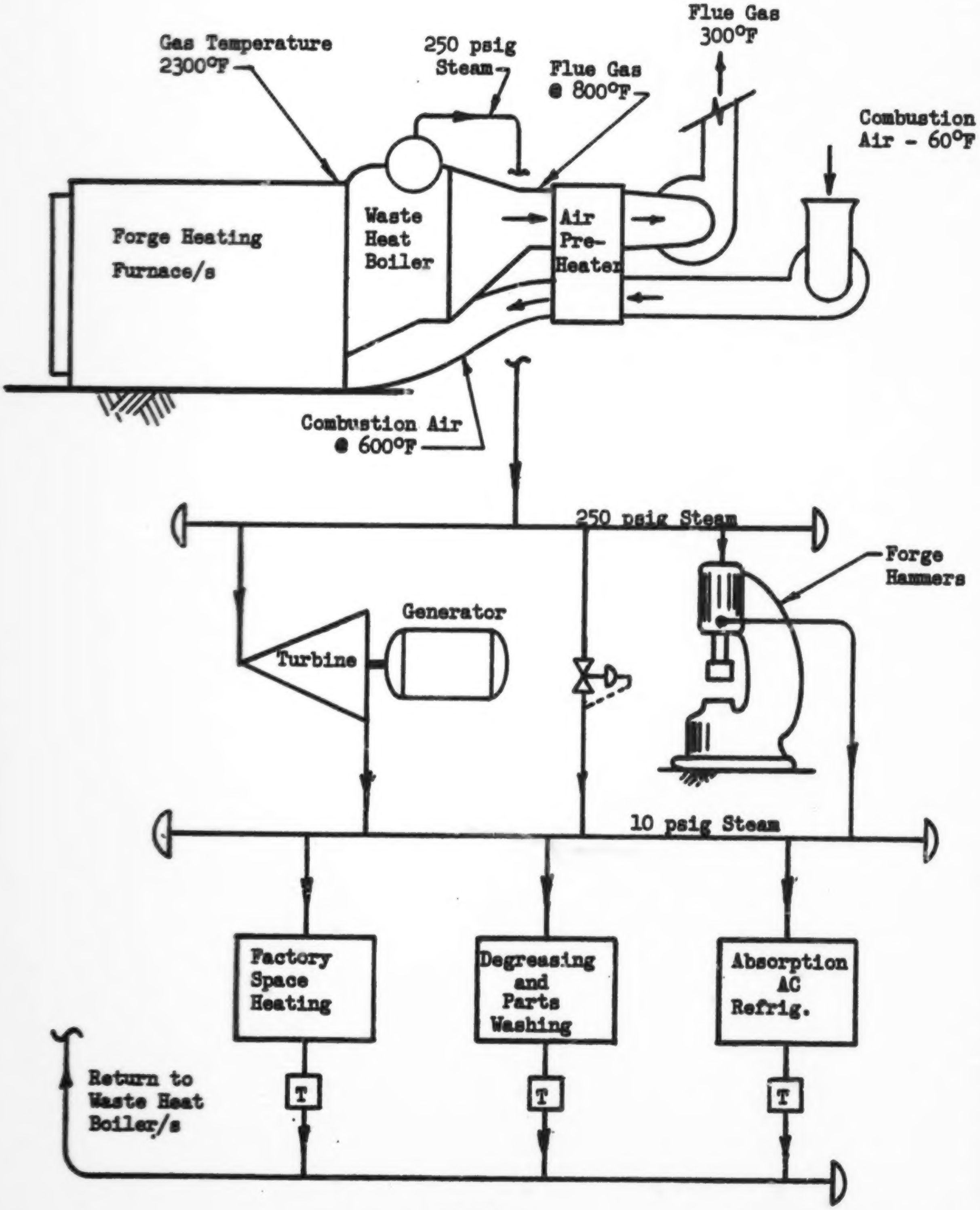
DISENGAGING DRUM
12.2" Hg A, 170°F

HOT PROCESS
WASTE WATER

7,500,000 LB/HR.
180°F

7,424,570 LB/HR. - 170°F

@ $0.04/KWH, POWER PRODUCED IS WORTH: $602,700/YEAR.



Gas Temperature 2300°F

250 psig Steam

Flue Gas @ 800°F

Flue Gas 300°F

Combustion Air - 60°F

Forge Heating Furnace/s

Waste Heat Boiler

Air Pre-Heater

Combustion Air @ 600°F

250 psig Steam

Forge Hammers

Generator

Turbine

10 psig Steam

Factory Space Heating

Degreasing and Parts Washing

Absorption AC Refrig.

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Return to Waste Heat Boiler/s